KARMAZIN & POPP

Design of a

150-Ton Rotary Tower Crane

Mechanical Engineering

B. S.

1911

DESIGN OF A 150-TON ROTARY TOWER CRANE

BY

JOHN KARMAZIN
PAUL FRED POPP

THESIS

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

John Karmazin and Paul Fred Popp

ENTITLED Design of a 150 Ton Rotary Tower Crane

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE

DEGREE OF Bachelor of Science in Mechanical Engineering

A. A. Leutwiler
Instructor in Charge.

APPROVED: G. A. Goodenough

Acting HEAD OF DEPARTMENT OF Mechanical Engineering

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ROTARY TOWER CRANE

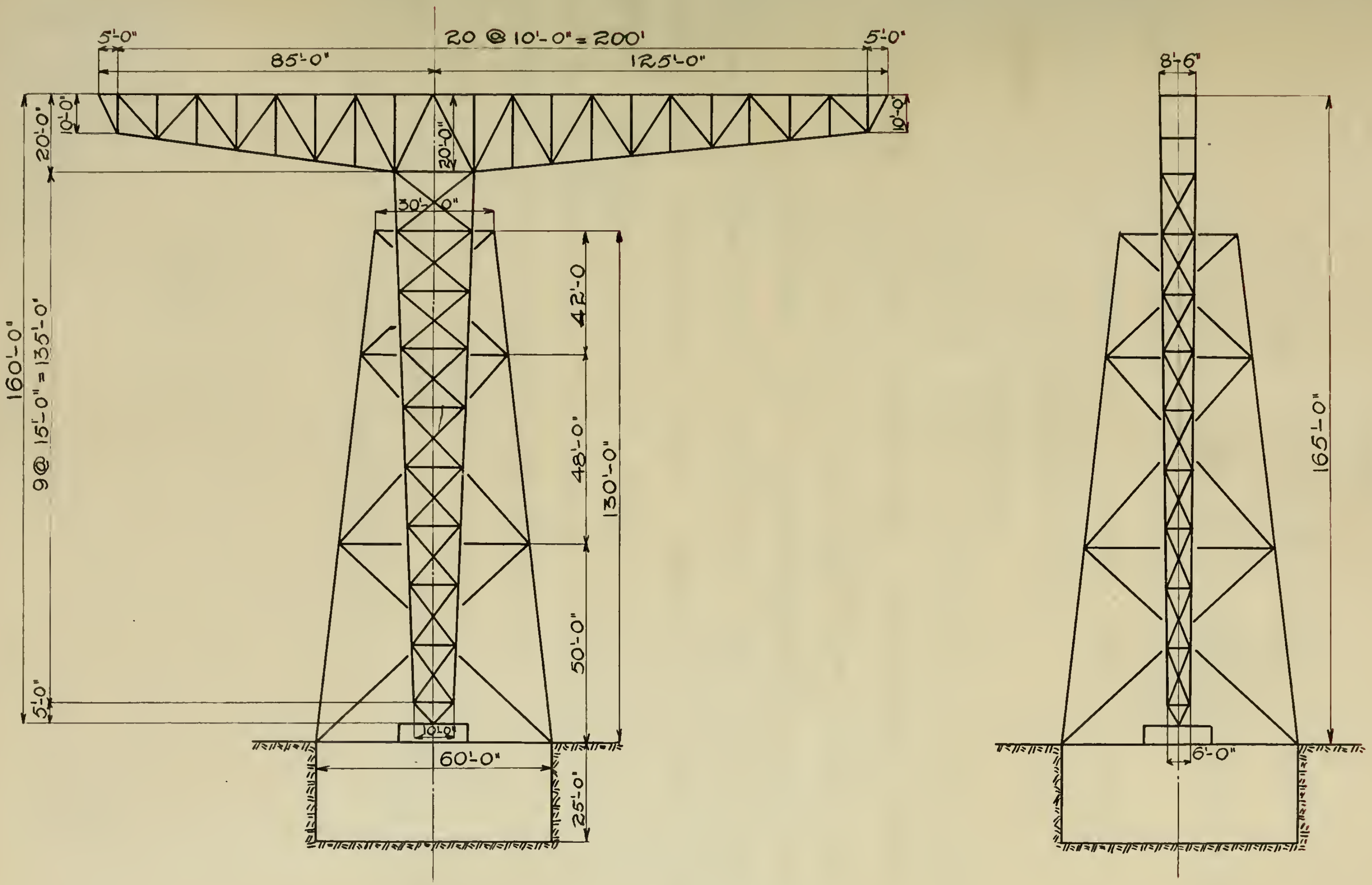
Introduction

The work of construction in ship building yards as well as the unloading of vessels at the docks requires the handling of very heavy masses. In order to convey these loads either on or off board the ships, special cranes have been designed for this purpose.

In the United States the cantilever traveling gantry crane is used almost exclusively for dock and ship building work while England and Germany have recently introduced a new type of crane, namely, the rotary tower crane. The first crane of this kind was built in Germany in 1903 with a capacity of 150 tons. Another crane of the same capacity and of German design was erected in England in 1906. It was very natural for English designers to attempt an improvement over the German crane, and this was accomplished the following year by building the tower columns perpendicular instead of tapering them inward from the bottom to the top. The center column of the German crane was replaced by a large roller path directly on top of the tower. The principal advantage of the English crane over the German is its lower cost particularly the cost of erection, since no false

work is required. The German crane on the other hand has a more pleasing appearance and shows greater stability and more careful design. The greater complexity of the German crane as a problem of design has been the chief reason for selecting that type as a thesis.

It is the purpose of this thesis to design a 150-ton rotary tower crane and present the subject not in complete detail but in such a manner as to show the method of analysing the various forces acting on such a structure and the determination of the stresses due to these forces.



20 @ 10'-0" = 200'
5'-0"
85'-0"
125'-0"
5'-0"
10'-0"
20'-0"
20'-0"
10'-0"
30'-0"
42'-0
48'-0"
50'-0"
130'-0"
160'-0"
9 @ 15'-0" = 135'-0'
5'-0"
10'-0"
60'-0"
25'-0"
8'-6"
165'-0"
6'-0"

Fig.1.

Chapter I.

Types of Rotary Tower Cranes

1. There are at present four types of rotary tower cranes in use, as follows:

1. Hammer type, in common use in Germany
    (a) Rectangular tower
    (b) Tripod tower
2. English type,- jib supported on top, roller path
3. Rotary column type- entire crane revolves
4. Traveling rotary tower type- German type

The hammer type of crane was chosen for this design for reasons stated in the introduction. The determination of the class of web bracing is of course a matter of choice with the engineer. Practice shows a wide variation in the type of truss used and it appears as if the choice is influenced by the personal desires of the designer more than by any other factor. In the case at hand the single Warren truss was used for both the lateral and vertical systems and the double Warren type with verticals for the outside and center towers.

Chapter II

General Dimensions

2. Determination of dimensions - The conditions under which a crane is to operate determine very largely its general dimensions. The height is fixed by the masts of the ships, and the radius at which the maximum load is to be lifted by practical considerations such as the width of the vessel and the range in which the crane can revolve efficiently under maximum load. In this design the length of the radius was arbitrarily chosen. The distance center to center of main trusses was made equal to the center distances between trolley wheels. The depth of the truss and panel length were so chosen as to procure an economical design of section and retain a pleasing appearance. The base dimensions of the outside tower were determined from the principles of stability. Fig. 1 gives the important dimensions of the structure a few of which are given below:

Available lift - - - - - - - - - - - - -120 feet

Radius with maximum load - - - - - - -110 feet

Radius with minimum load - - - - - - -110 feet

3. Loads -

Main lift - - - - - 150 tons

Auxiliary- - - - - 60 tons

Wind on upper chord 250 pounds per lineal foot

Wind on lower chord 250 pounds per lineal foot

Wind on the tower - 100 pounds per vertical lineal foot

The weight of the trolley was computed after the design was made and will be given in tabulated form in the following pages. The dead load of the tower and revolving jib had to be assumed, however the assumptions made were based upon the weights of cranes of the same capacity as the one under consideration and are fairly accurate. The weight of the revolving jib of 210 tons was evenly distributed over the lower panel points as shown on the stress sheet Fig. 36 which produced a load of 20000 pounds at each panel point. None of this dead load was applied to the panel points of the center column because the method of distributing all the load over the truss gives slightly larger stresses in the tower columns and avoids unnecessary refinement.

The dead load of the outside tower of 100 tons was distributed proportionately among the panel points with increasing loads from top to bottom. Stress sheet Fig. 38 shows the magnitude and application of the loads.

4. Speeds- Speed of hoisting (150 tons)- 3 feet per minute

Speed of hoisting (60 tons)-18 feet per minute

Racking speed - - - - - - - 25 feet per minute

Slewing speed - - - - - - - One revolution in ten minutes

Chapter III

Specifications

5. Trolley - The trolley will be of the steel frame type of rigid construction. All bearings shall have ample bearing area with bronze bushings on all main shafts.

Drum - The drum shall be machine grooved and of ample size for the rope used and to permit of full hoist without overlapping of ropes.

Brakes - The hoist shall be equipped with two brakes, a mechanical load brake and an automatic solenoid electrical brake. Both brakes shall be capable of holding the load independently.

Gears - All gears except the drum and pinion are cut from the solid from good gray iron or cast steel. Pinions are cut from forged steel blanks.

Shafting - All large shafts to be forged steel, small shafts of cold rolled.

Hook and Block - To be of the Hess-Bright ball-bearing type carrying a heavy hook of refined steel. Block sheaves to be of ample diameter for the rope used and to have turned grooves. The sheaves to be fitted with bronze bushings.

Factor of Safety - All parts of the trolley to be designed so that the maximum unit stresses shall not exceed the following values:

Tension - - - - - - - - - - - - 16000 lbs.per sq.in.

Compression - - - - - - - - - 16000 " " " "

Shear - - - - - - - - - - - - -	10000 lbs. per sq.in.	
Bearing (journal) - - - - -	1500 to 3000 " "	
Bearing (stationary surface)	24000 lbs. per sq.in.	

6. Structure - All structural steel must conform to manufacturers' standard specifications.

7. Unit Stresses and Proportion of Parts - All parts of the structure shall be so proportioned that the sum of the maximum stresses produced by the loads shall not exceed the following amounts in pounds per s quare inch.

Axial tension on net section	16000
Axial compression on gross section of columns	16000-70 l/r
l = length of member in inches	
r = least radius of gyration in inches	
Direct compression	16000
Bending on extreme fibers	16000
Shearing, shop rivets and pins,	12000
field rivets and turned bolts	10000
Bearing, shop rivets and pins	24000
field rivets and turned bolts	20000

Alternate Stresses - Members subject to alternate tension and compression shall be proportioned for stresses giving the largest section. If the alternate stresses occur in succession during the passage of the trolley from outer to inner positions each stress shall be increased by 50 percent of the smaller. The connections shall in all cases be proportioned for

the sum of the stresses. For the details of design any standard bridge specifications may be followed preferably those of the "American Railway Engineering and Maintenance of Way Association".

Chapter IV

The Trolley

8. Design of 150 Ton Hook

The double hook type was selected since it is better adapted to heavy loads.

Load 300,000 pounds

Unit stress tension 16000 lb. per sq.in.

Area required at bottom of thread

$$= \frac{300,000}{16,000} = 18.75 \text{ sq.in.}$$

Dia. = 5"

Use U.S.S. Thread 6 threads per inch

Dia. at root = 5.0 inches

Outside dia. = 5.25 inches

The maximum stress in a double crane hook according to Bach is:

$$S = \frac{Q \sin\alpha}{2\,A} - \frac{Q\,x}{2\,A\,r} + \frac{Q\,x}{2\,z\,A\,r}\,\frac{a}{r-a} \quad - - - - \quad (1)$$

S = unit stress lb. per sq.inch

Q = total load in lb.

α = angle section taken makes with the vertical

A = area of section

x = distance to center of gravity of section

r = radius of curvature of gravity axis

a = distance to extreme fiber

b = one half of the short diameter of the ellipse

$$z = -\frac{1}{A}\int \frac{\eta}{r+\eta}\, dA$$

z for an elliptical section has the following value

$$z = \frac{1}{4}\left(\frac{a}{r}\right)^2 + \frac{1}{8}\left(\frac{a}{r}\right)^4 + \frac{5}{64}\left(\frac{a}{r}\right)^6 + - - \qquad (2)$$

$\sin \alpha = .32$

$A = \pi a\ b = \pi 6 \times 3 = 56.5$ sq.inches

$$z = \frac{1}{4}\left(\frac{6}{18}\right)^2 + \frac{1}{8}\left(\frac{6}{18}\right)^4 + \frac{5}{64}\left(\frac{6}{18}\right)^6$$

$$= .026$$

Substituting these values in (1) above, we have

$$S = \frac{150000 \times .32}{56.5} - \frac{150000 \times 5.75}{56.5 \times 18}$$

$$+ \frac{150000 \times 5.75}{.026 \times 56.5 \times 18}\ \frac{6}{18-6}$$

$$= \pm\ 16300 \text{ lb. per sq.in.}$$

Fig. 2 shows in detail the dimensions of the hook finally adopted for 150 ton capacity

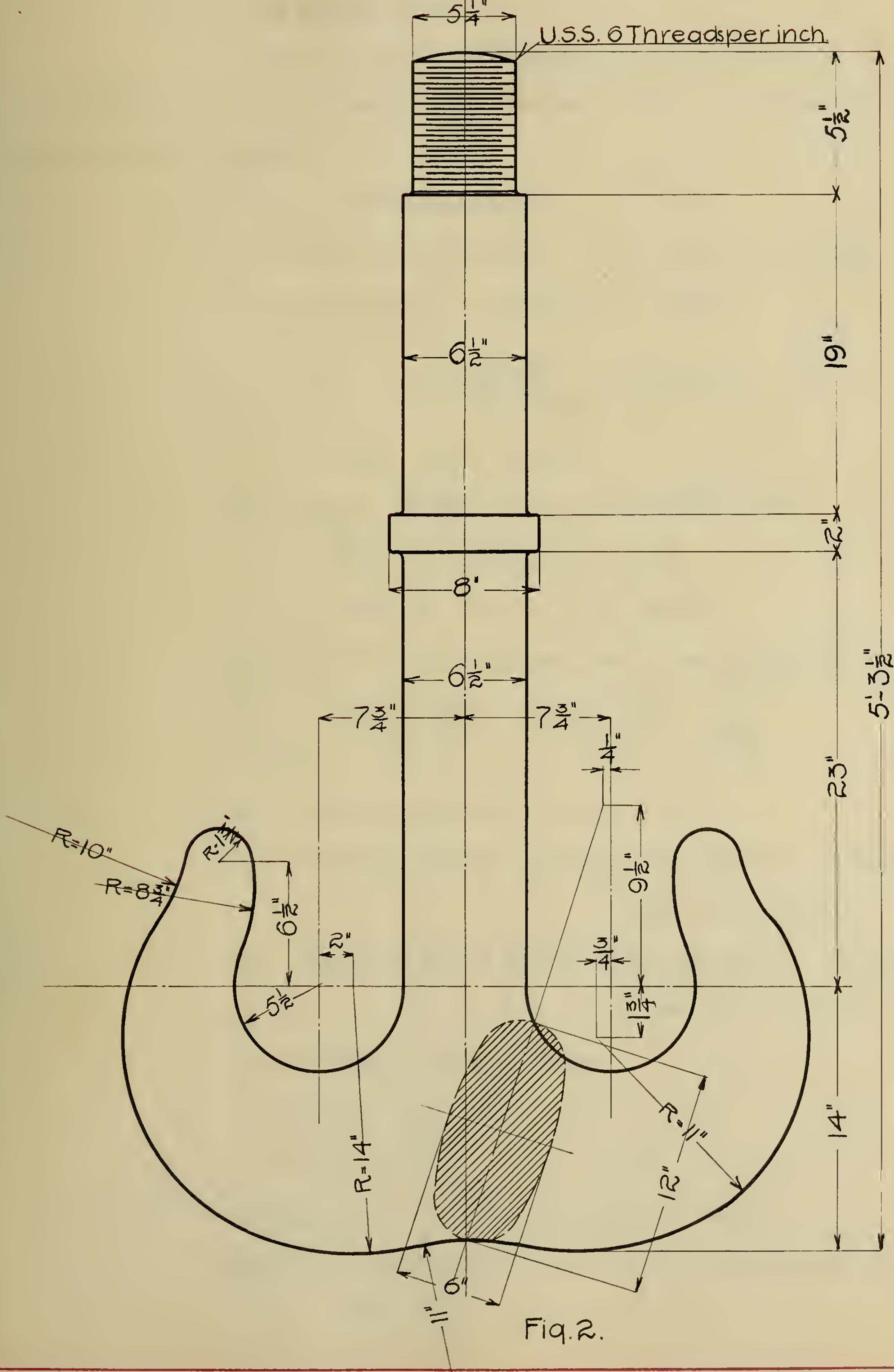

5 1/4"
U.S.S. 6 Threads per inch.
5 1/2"
6 1/2"
19"
2"
8"
6 1/2"
5'-3 1/2"
7 3/4"
7 3/4"
1/4"
23"
R=10"
R-1 1/4"
R=8 3/4"
6 1/2"
9 1/2"
2"
3/4"
5 1/2"
1 3/4"
R=11"
14"
R=14"
12"
6"
11"

Fig. 2.

9. <u>60 Ton Crane Hook</u>

The type of hook selected for this capacity is not the same as shown in Fig. 2, but of the ordinar y form in common use in this country

Load 120,000 pounds

Unit tensile Strength 16000 lb. per sq.in.

Area required at bottom of thread

$$= \frac{120,000}{16,000} = 7.5 \text{ sq.in.}$$

Dia. = 3.1 inches

Use U.S.S. Thread 6 threads per inch

Dia. at root = 3.1 inch

Outside dia. = 3.35 inch

Maximum stress in crane hook according to Bach is

$$S = \frac{Q}{A} - \frac{Q(a + e_2)}{A\,r} - \frac{Q(a + e_2)}{z\,A\,r}\;\frac{\eta}{r + \eta} \qquad (3)$$

a = distance shown on sketch

e_2= distance from center of gravity to extreme fiber in tension

e_1= distance from center of gravity to extreme fiber in compression

η = distance to any fiber

$$z = -\frac{1}{A}\int \frac{\eta}{r + \eta}\, dA$$

Other symbols as in previous article

The following are values for a trapezoidal section see Fig. 3.

$$A = \frac{b + b_1}{2} h$$

$$A = \frac{3\ 1/4 + 9\ 3/4}{2}\ 10$$

$$= 65 \text{ sq.inches}$$

$$e_2 = \frac{h}{3} \frac{b + 2 b_1}{b + b_1}$$

$$e_1 = h - e_2$$

$$r = a + e_2$$

$$h = 2 a$$

$$b = 3 b_1$$

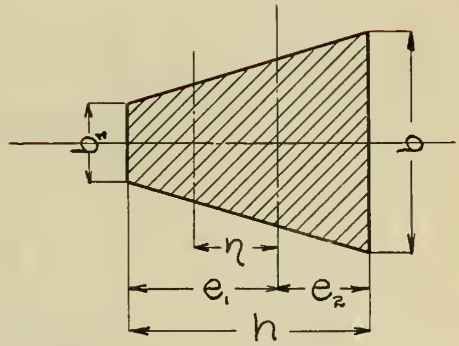


Fig. 3.

$$z = -1 + \frac{2 r}{(b + b_1)h} \left\{ \left[b_1 + \frac{b-b_1}{h}(e_1+r) \right] \log_e \frac{r+e_1}{r-e_2} - (b-b_1) \right\} \quad (4)$$

Taking above proportions we have:

$e_2 = 5/6\ a \qquad r = 11/6\ a$

and the expression for z reduces to $1/z = 10.27$

Substituting the above values in equation (3), we have:

$$S = -10.27 \frac{Q}{A} \frac{\eta}{11/6\ a + \eta} \quad - - - - (5)$$

For $\eta = -e_2$ equation (5) reduces to

$$S_t = + 8.56 \frac{Q}{A} \quad - - - - - - - - - - - (6)$$

$$S_c = - 3.99 \frac{Q}{A} \quad - - - - - - - - - - - (7)$$

For the 60 ton hook the line of action of Q is 5 inches from inside of hook.

$b_1 = 3\ 1/4$ inches; $b = 9\ 3/4$ inches; $h = 10$ inches;

$A = 65$ sq.inches

$$S_t = + 8.56 \frac{120000}{65} = + 15900 \text{ lb. per square inch}$$

$$S_c = - 3.99 \frac{120000}{65} = - 7350 \text{ lb. per square inch}$$

The dimensions finally selected for this capacity are shown in Fig. 4.

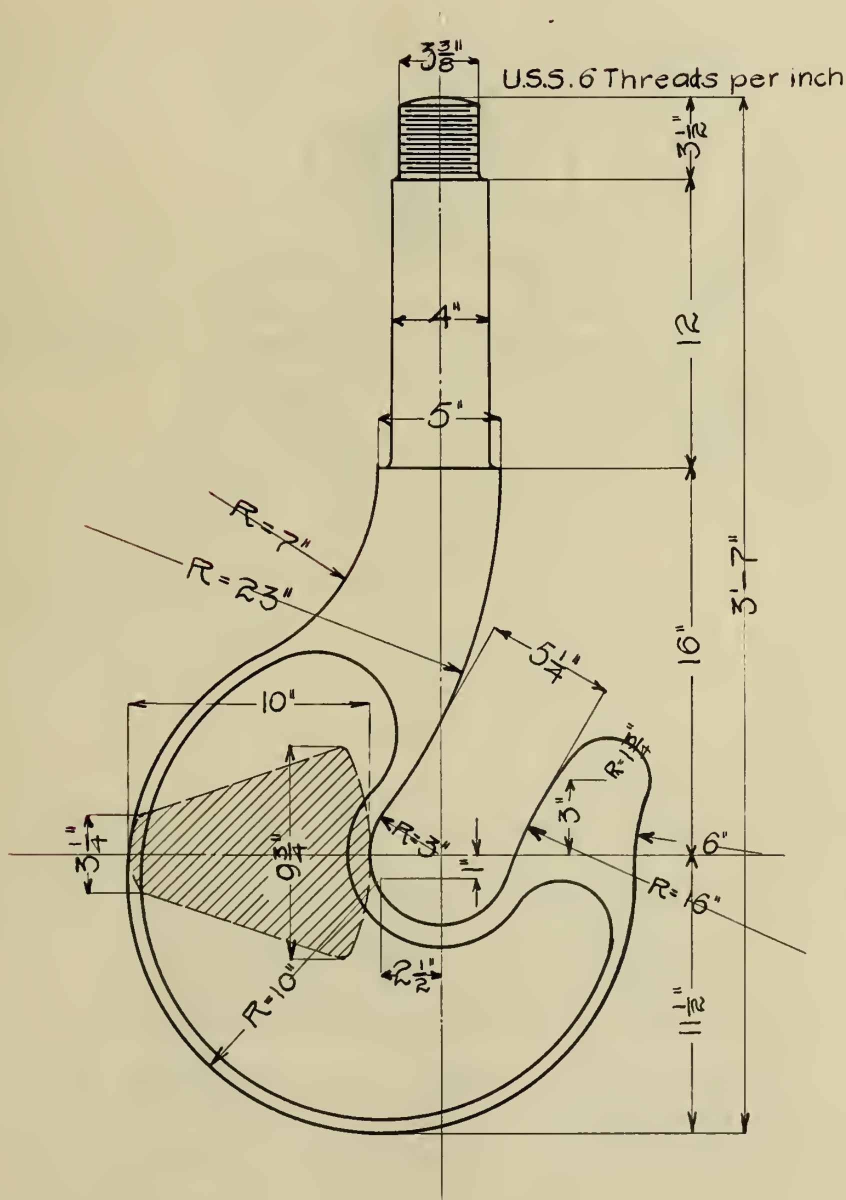


Fig. 4.

10. <u>Design of Hook Thrust Bearing</u>

The type of thrust bearing selected is shown in Fig. 5. The dimensions shown in this figure apply to the bearing selected for the 150 ton hook, and were obtained from the Hess-Bright Co. catalog. It is known as their "Medium 1100 Series"

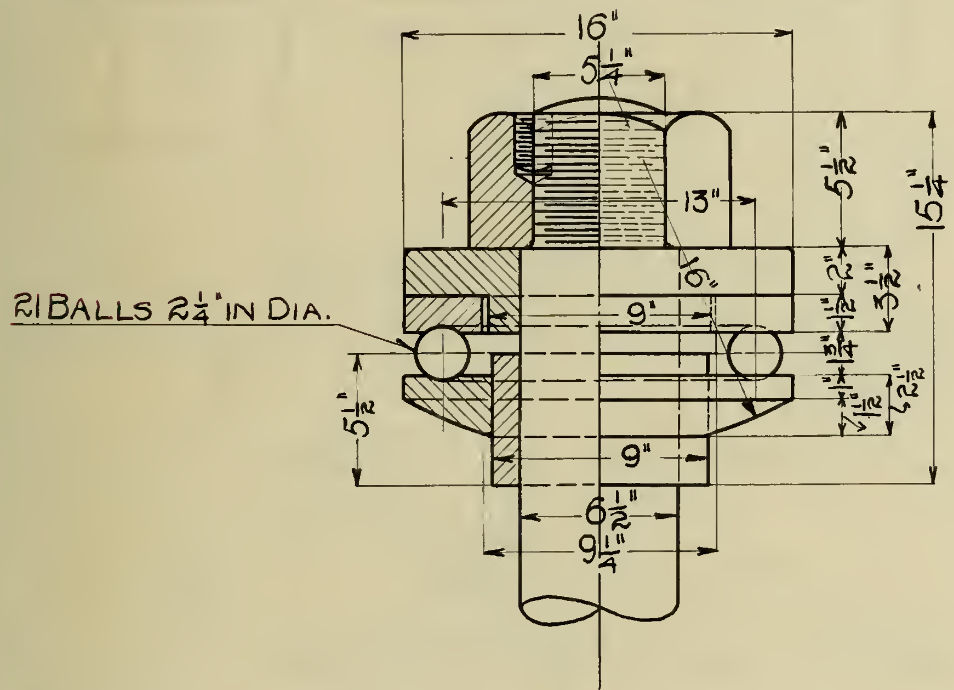


Fig. 5.

For the 60 ton hook a bearing similar to the one shown in Fig. 5 will be used, and the following are the dimensions:

Inside diameter	9 1/4 inches
Outside diameter	13 9/16 inches
Thickness of height	3 3/4 inches
Diameter of ball	1 5/8 inches
No. of balls	34

11. <u>Size of Sheave pin</u> for 150 Tons Load

The loading of the sheave pin for the 150 ton block is shown in Fig. 6.

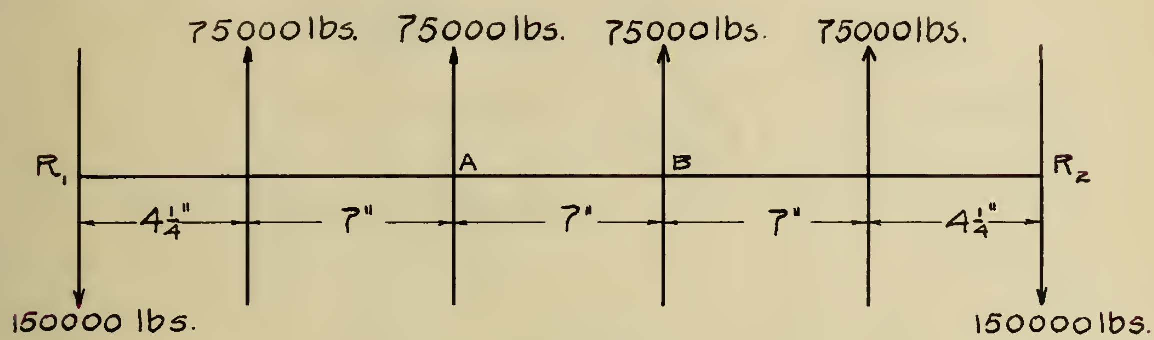


Fig. 6.

Take moments at A, we have

$$-150000 \times 11\ 1/4 \text{ inches} + 75000 \times 7 =$$

$$1,165,000 \text{ in.lb.}$$

Allowable unit stress 16000 lb. per sq.in.

$$\frac{1,165,000}{16000} = \frac{I}{c} = 72.75$$

Hence the diameter of pin = 9 inches

12. <u>Size of Sheave pin</u> for 60 Tons Load

Fig. 7 shows the application of the loads coming upon the sheave pin of the 60 ton block

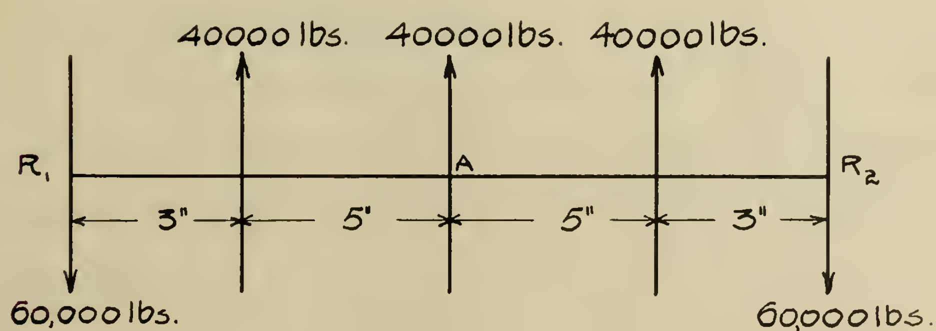


Fig. 7.

Take moments at A, we have

$$-60000 \times 15 + 40000 \times 5 = 600{,}000 \text{ in.lb.}$$

Allowable unit stress 16000

$$\frac{600{,}000}{16{,}000} = \frac{I}{c} = 37.5$$

Hence the diameter of pin = 7 1/4 inches

13. Design of Sheaves

150 Ton Block

Diameter of sheaves 3 ft.6 in.

Length of hub 7 in.

Allowable unit bearing pressure 2000 lbs.

Actual bearing pressure = 75000 ÷ (7 x 9) = 1900 lb. per sq.in.

60 Ton Block

Diameter of sheaves 22 in.

Length of hub 5 in.

Allowable unit bearing pressure 2000 lbs.

Actual bearing pressure = 40000 ÷ (5 x 7 1/4) = 1100 lbs. per sq.in.

14. Design of Block Pin

The load acting upon the block pin for both the 150 ton and 60 ton capacities will be assumed as concentrated at the center, as shown in Fig. 8, which figure applies to the 150 ton block.

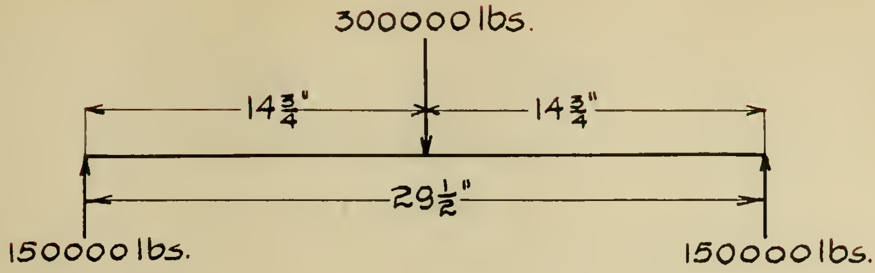


Fig. 8.

The maximum bending moment

$M = 150{,}000 \times 14\ 3/4 = 2{,}212{,}000$ lb.in.

$$\frac{2{,}212{,}000}{16000} = \frac{I}{c} = 830$$

Diameter of pin = 9 3/8 inches at center

Size of pin in bearings

Allowable unit shear 10000 lb.

Actual shear = 150,000 lb.

$$\frac{150{,}000}{10{,}000} = 15 \text{ sq.in. required}$$

Diameter = 4 1/2 inches

<u>60 Tons</u>

By a similar line of procedure the following dimensions of the 60 ton block pin are f ound:

Diameter of Pin = 5 5/8 inches at center

Diameter of Pin in bearings = 2 7/8 in.

15. <u>Size of Block Plates</u>

150 Tons

Allowable unit bearing s tress 20000 lbs.

$P = 300{,}000$

$$\text{Area} = \frac{300{,}000}{20{,}000} = 15 \text{ sq.in.}$$

Diameter of pin 4 1/2 in.

t = plate thickness

2t x 4 1/2 = 15

t = 1.67 say 1 3/4 in.

<u>60 Tons</u>

Thickness of Plate = 1 1/8 in.

16. <u>Design of Trolley Block</u>

<u>150 Tons - Trolley Sheave Pin</u>

In Fig. 9 are shown all the loads acting upon this pin, and they will be considered as concentrated at the center of the sheave bearings.

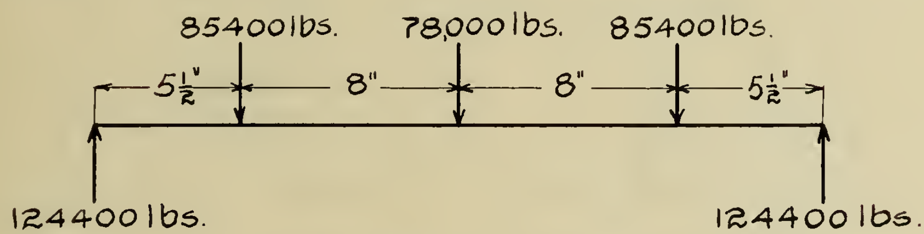


Fig. 9.

Taking moments at A, we have

124,400 x 13 1/2 - 85400 x 8 = 996,000 in.lb.

$$\frac{996,000}{16,000} = \frac{I}{c} = 62.3$$

Diameter = 8 1/2 in.

Bearing pressure on sheave pin

$$\frac{85,400}{68} = 1250 \text{ lb. per sq.in.}$$

Allowable = 2000

<u>Size of Plate</u>

Allowable bearing pressure 20,000 lb. per sq.in.

$$\text{Area required} = \frac{124,400}{20,000} = 6.22 \text{ sq.in.}$$

t = thickness of plate

$8\ 1/2\ t = 6.22$

$t = 3/4$ in.

<u>60 Tons - Trolley Sheave Pin</u>

The loading of the pin is shown in Fig. 10, and treating it in a manner similar to that given above, we find the maximum bending moment as

$$M = 42800 \times 3 = 128,400 \text{ lb.in.}$$

$$\frac{128,400}{16,000} = \frac{I}{c} = 8$$

Diameter = 4 15/16 in.

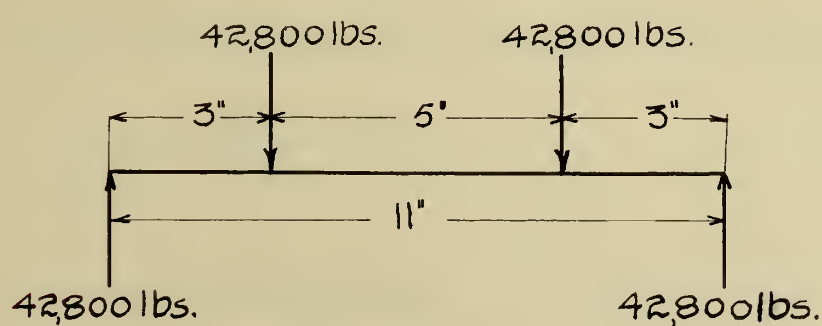


Fig. 10.

Thickness of plate = 1/2 in.

17. Rope Tensions - The arrangement of the hoisting rope for the 150 ton block is shown diagrammatically in Fig. 11. Assume the coefficient k as 1.06. The weight of the hook and block is 12000 pounds, hence the total load to be raised is 312000 pounds.

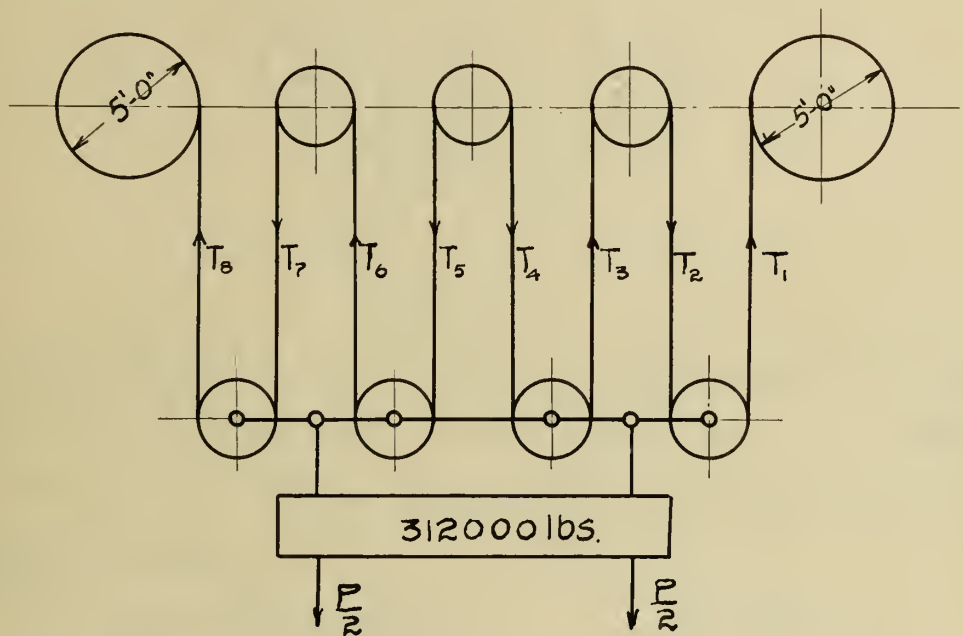


Fig. 11.

From the theory of rope stiffness we have the following equation:-

$$T_4 = \frac{156000\ (k-1)}{k^4-1}$$

$$= \frac{156000\ (.06)}{.264}$$

$$= 35450 \text{ pounds}$$

$$T_1 = T_4k^3 = 35450 \times 1.06^3 = 42300 \text{ pounds}$$

$$T_1 = T_8 = 42300 \text{ pounds}$$

The value 42300 fixes the size of rope used in this case.

For the 60-ton lift the arrangement of the hoisting rope is shown in Fig. 12.

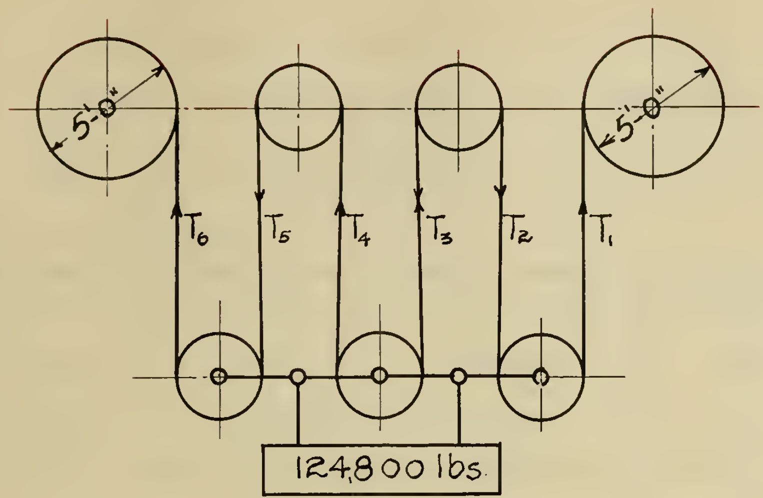


Fig. 12.

In this case the weight of the block is 4800 pounds which makes the load to be raised as 124800. The value of k will be assumed the same as above. By a similar line of reasoning used above. we find the following values:

$$T_3 = 19600 \text{ pounds}$$

$$T_6 = T_1 = 22200 \text{ pounds}$$

The rope must be made large enough to resist this pull of 22200 pounds.

18. Size of Rope - 150-Ton Block

From Art. 17 the rope tensions T_1 and T_8 were found to be 42300 pounds, which requires the use of a 1 3/4 inch plough steel rope. The following table gives additional data applying to this rope.

Diameter of Rope inches	Weight per Foot	Average safe working stress- lb.	Weight in lb.	Length in ft.
1 3/4	4.85	51200	6500	1333

60-Ton Block- From Art. 17 the rope tensions T_1 and T_6 were found to be 22200 pounds. This requires a 1 1/4 inch plough steel rope. The following table gives further data pertaining to the rope selected.

Diameter of rope inches	Weight per foot	Average safe working load	Weight in lb.	Length in ft.
1 1/4	2.45	26800	2475	1010

19. Diameter of Drums - 150-Ton. Practice shows that the diameter of a crane trolley drum is about 30 times the rope diameter. Since the rope is 1 3/4 inches a drum 5 feet in diameter is chosen. The amount of rope to be wound on the drum is 150 x 4 or 600 feet.

Pitch of grooves = 2 in.

Number of turns $n = 600/5\pi = 38.2$

Length of drum assuming three extra turns is 7 feet.

60-Ton - For the 60 ton load a much smaller rope is required and consequently a smaller drum, but for practical reasons this drum will be made the same diameter as the 150-ton drum. The amount of rope to be wound on the drum is 150 x 3 or 450 feet.

Pitch of grooves = 1.5 in,

Again assuming three extra turns the length is 4 feet.

20. Horse-power of Motors - The size of the motors required is determined from the load and speed with which it is to be hoisted. The following gives the method of procedure:

Efficiency of rope = 95 percent

Efficiency of each pair of gears = 96 percent

Efficiency of worm gear = 45 percent

Efficiency of motor = 80 percent

Speed = 3 feet per minute

Combined efficiency of rope and gears = 41.5 percent

$$H.P. = \frac{42300 \times 12}{.415 \times 33000} = 37.1$$

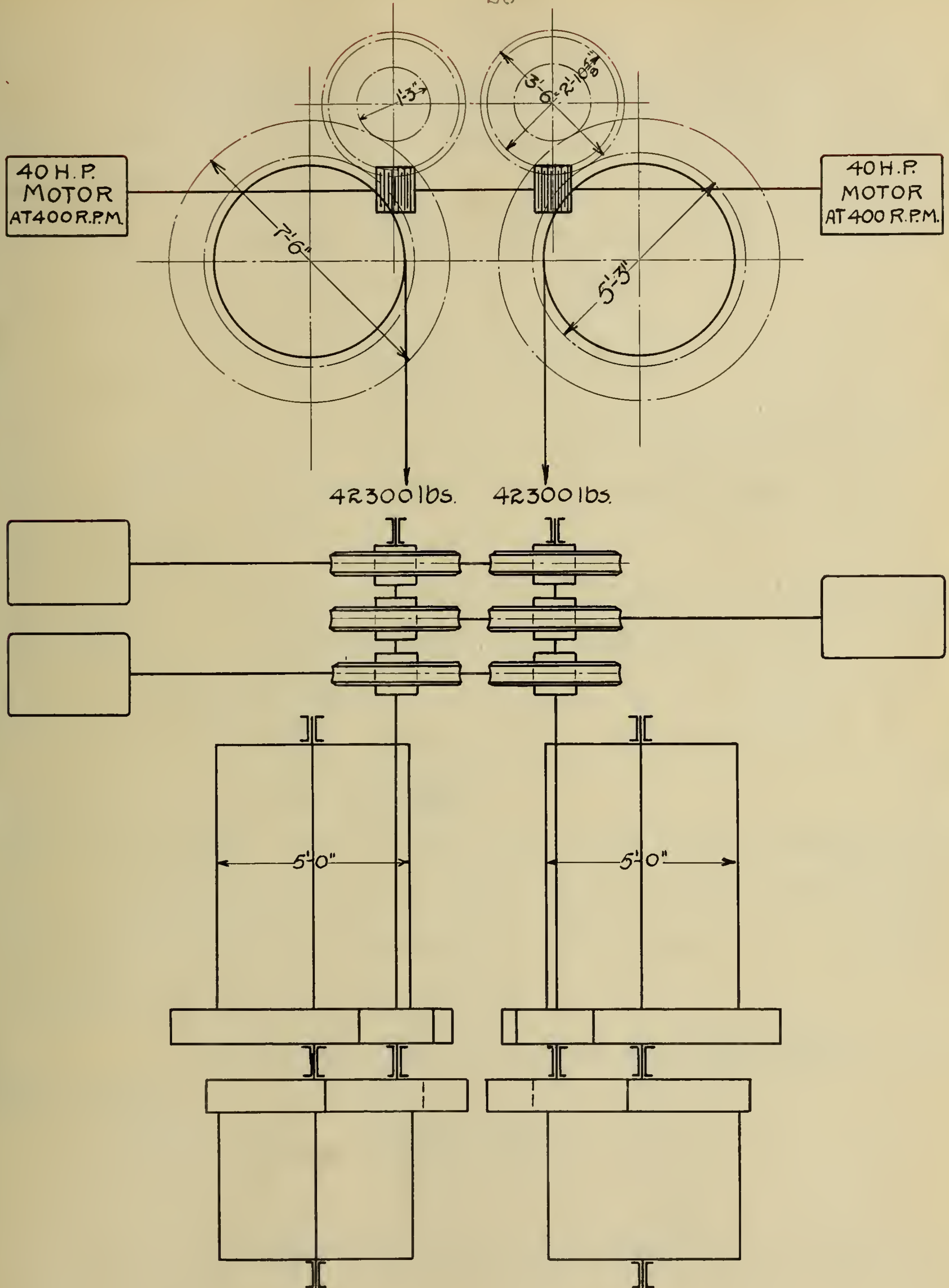
$$\text{Actual H.P.} = \frac{37.1}{.8} = 46.40$$

Each motor must deliver 2/3 x 46.40 = 30.9 H.P.

The nearest commercial size of motor that can be used is one rated at 40 H.P. at 400 R.P.M. Assuming the speed of the rope as 12 feet per minute, the gear ratio between the motors and thedrum will have to be

$$\frac{400 \times 15.7}{12} = 522$$

Upon careful consideration as to the system of gearing best suited for driving the drum, the German method of worm gear drive will be selected. It has the advantage of being a much lighter and simpler mechanism than the spur gear drive which for such heavy loads would require exceedingly massive gears. Fig. 13 shows a diagrammatic arrangement of the mechanism. Assume the diameter of the gear on the drum as 7 ft. 6 in. and that of the pinion 15 in.



40 H.P. MOTOR AT 400 R.P.M.
40 H.P. MOTOR AT 400 R.P.M.
1'-3"
3'-6"
2'-10 5/8"
7'-6"
5'-3"
42300 lbs.
42300 lbs.
5'-0"
5'-0"

Fig. 13.

21. Design of Drum Gear and Pinion - 150-Ton. The load coming upon the teeth of the main gear and pinion in this case is as follows:

$$\text{Load } W = \frac{42300 \times 30}{.96 \times 45} = 29400 \text{ pounds}$$

The velocity of the gear teeth is

$$\frac{12}{15.7} \times 7.5 \times 3.14$$

or approximately 18 feet per minute. As a trial substitution in the Lewis formula, assume the following:

$p = 1$; $p' = 3.142$; $y = 0.072$; $n = 15$; $D = 15$ in.

S for steel casting = 19000; $f = 7$ in.

$$W = Sp'fy$$

$$29400 = 19000 \times 3.142 \times 7 \times 0.072$$

$$29400 \neq 30080 \text{ lb.}$$

For all practical purposes this is a close enough agreement, hence the above dimensions will be used. Making the gear of the same material it will be unnecessary to investigate it for strength as the pinion is always the weaker.

Worm Gear- The load coming upon the worm gear is as follows:

$$\text{Load } W = \frac{29400 \times 7.5}{17.31 \times 3} = 4240 \text{ lb.}$$

$$\text{Velocity} = \frac{3.14 \times 34.62 \times 4.6}{12} = 41.7 \text{ ft.per min.}$$

Assume $y = 0.11$; $f = 2\ 3/4$ in.; $p' = 1.25$ in.

S for bronze = 11500; substituting these values in the Lewis formula we get:

$$4240 = 11500 \times 1.25 \times 2.75 \times 0.11$$
$$\neq 4350$$

The agreement is close enough hence the dimensions assumed will be used. The diameter of the gear is 34 5/8 inches and the number of teeth 87.

22. <u>Drum Gear and Pinion</u> - 60-Ton. For this lighter load assume a rope speed of 54 feet per minute. The gear ratio between the motors and the drum will necessarily be

$$\frac{400 \times 15.7}{54} = 116$$

Assume the diameter of the gear on the drum as 5 ft. 3 in., that of the pinion as 42 inches, and the load coming upon the teeth of the gear and pinion is as follows:

$$\text{Load } W = \frac{22200 \times 30}{.96 \times 31.5} = 22000 \text{ lb.}$$

As a trial substitution in the Lewis formula assume the following:

$$p' = 1.57;\ p = 2;\ n = 84;\ f = 6.75;\ y = .113$$

S for steel casting = 18500 lb.

$$22000 = 18500 \times .113 \times 6.75 \times 1.57$$
$$\neq 22200$$

The agreement is sufficient, hence the assumed dimensions will be used. The diameter of the pinion is 42 inches and the number of teeth 84.

The gear is made of the same material which will make an investigation for its strength unnecessary because the pinion is always the weaker.

23. Design of Worm - The motor is capable of transmitting 40 H.P. at 400 R.P.M., hence the twisting moment coming upon the worm shaft is

$$P_p = \frac{63030\ H}{n}$$

$$= 6303 \text{ lb. inches}$$

Assume $\alpha = 4^\circ$ $\tan\alpha = .0699$

$$u' = .05\ \sqrt{1 + .072 \times .989} = 0.052$$

Worm 1 1/4" pitch 75° involute teeth

Force transmitted to gear at pitch line

$$F = \frac{29400 \times 7.5''}{17.30 \times 3} = 4250\#$$

P. Diameter of Worm gear = 34.62 "

$$\text{P. Diameter of Worm} = \frac{1.25}{3.14 \times .07} = 5.7''$$

$$P = W\left(\frac{p' + 2\mu'\pi r}{2\pi r - \mu' p'}\right)$$

$$= 566 \text{ lb.}$$

$$P_o = 298$$

$$= \frac{P_o}{P} = \frac{298}{566} = 52.5\%$$

$$S = W\left(\frac{\tan\beta}{1 - \mu'\tan\alpha}\right)$$

$$= 4665\frac{.268}{1 - .052 \times .07} = 1140 \text{ lb.}$$

24. Design of Worm Shaft - A diagram of the forces acting on the worm shaft is shown in Fig. 14. The horizontal thrust is neutralized by the right and left hand worms.

$$R_1 = 1140 \text{ lb.}$$

$$R_a = \sqrt{\overline{1140}^2 + \overline{566}^2}$$

$$= 1210$$

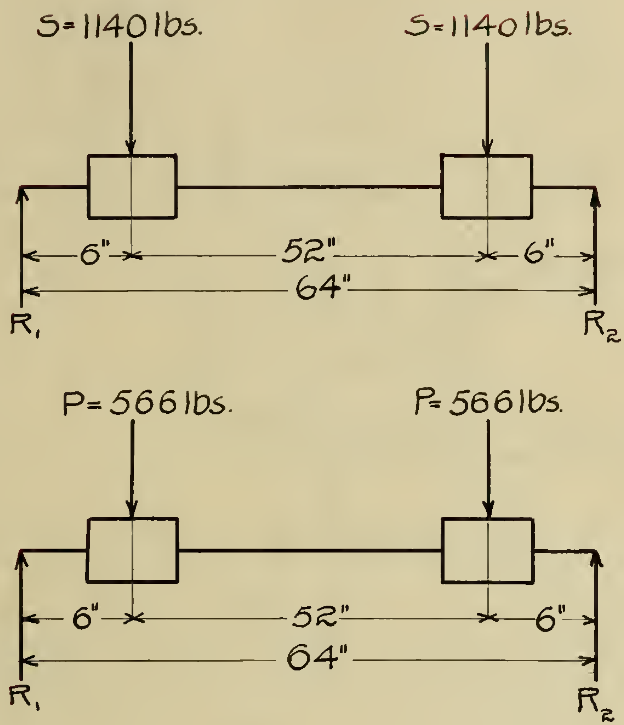


Fig. 14.

The maximum bending moment is

$$1210 \times 6 = 7260 \text{ lb.in.}$$

Substituting in Guest's Law, we have

$$M_e = \sqrt{\overline{6303}^2 + \overline{7260}^2} = 9620 \text{ lb.in.}$$

$$\frac{9620}{4000} = 2.4 = \frac{I}{c}$$

Diameter of shaft = 2 7/8 in.

25. Design of Worm Gear Shaft - The forces acting upon the worm gear shaft are shown in Fig. 15.

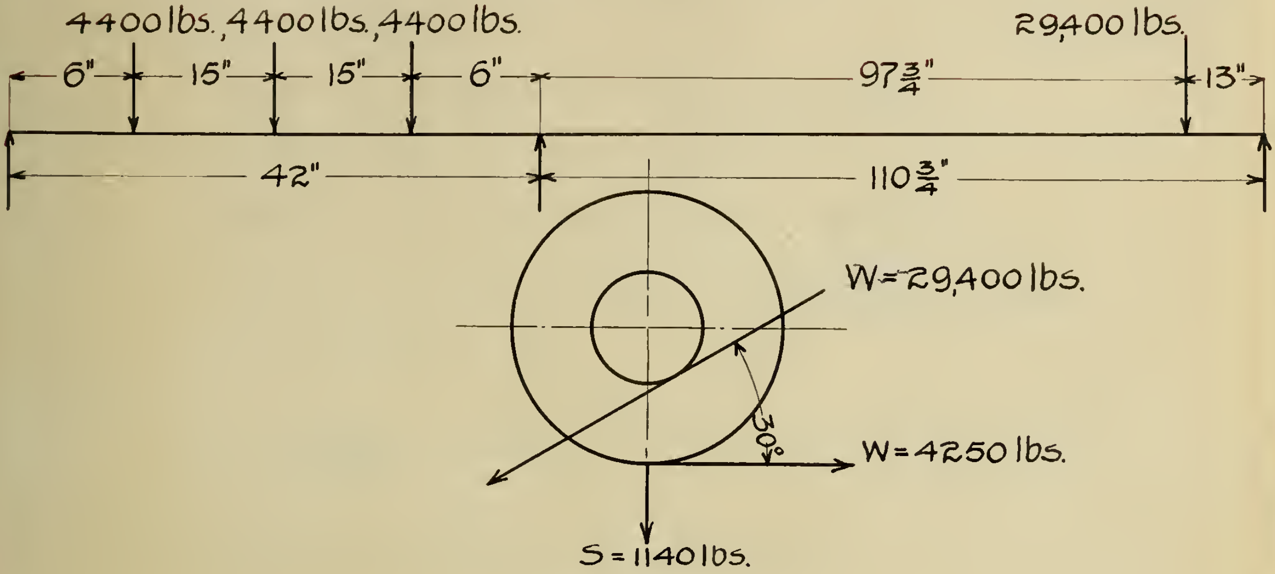


Fig. 15.

The shaft is continuous over three supports and the reactions are determined according to the theorem of three moments as follows:

$$M_1 l_1 + 2M_2(l_1 + l_2) + 3M_3 l_2 = - P_1 l_1^2(k_1 - k_1^3)$$
$$- P_2 l_2^2(2k_2 - 3k_2^2 + k_2^3)$$

$$2\ M_2(42 + 110.75) = -4400 \times \overline{42}^2\ (.143 - .\overline{143}^3)$$

$$M_2 = -3560$$

Since there are three forces acting there will be three values of k, hence by applying the above theorem three times the total moment at the second support is

$$-3560 - 9550 - 5775 = -18885$$

$$42\ R_1 - 4400(6 - 21 - 36) = -18885$$

$$R_1 = 6150$$

Maximum bending moment is 6150 x 6 = + 36900 in.lb. By a similar method of procedure the reaction and moment due to the load of

29400 pounds are found to be

$$2\ M_2(42 + 110.75) = -29400 \times \overline{110.75}^2(2 \times .88 - 3 \times \overline{.88}^2 + \overline{.88}^3)$$

$M_2 = -130000$

$110.75\ R_3 - 29400 \times 97.75 = -130000$

$R_3 = +24800$ lb.

Bending moment is 24800 x 13 = 322000 lb.in.

Twisting moment is 29400 x 75 = 222000 lb.in.

$M_e = 392000$ lb.in.

$$\frac{I}{c} = \frac{392000}{16000} = 24.5$$

Diameter of shaft = 6 1/4 inches

26. <u>Design of Drum</u> - The drum must be strong enough to withstand the bending moment due to the forces acting shown in Fig. 16 and in addition those due to its own weight; furthermore it must be stiff enough so that deflection will not exceed 0.01 of an inch. A thickness of shell will be assumed and investigated for maximum unit stress. The fiber stress found to be 168 pounds per square inch is very low but further investigation for deflection as shown in the following calculations reconciles the assumed thickness.

Max. M_B takes place with load at center

$M_B = Wl/4$

$$= \frac{42300 \times 84}{4}$$

$= 890,000$ lb.in.

$M_T = 42300 \times 30$

$= 1,269,000$ lb.in.

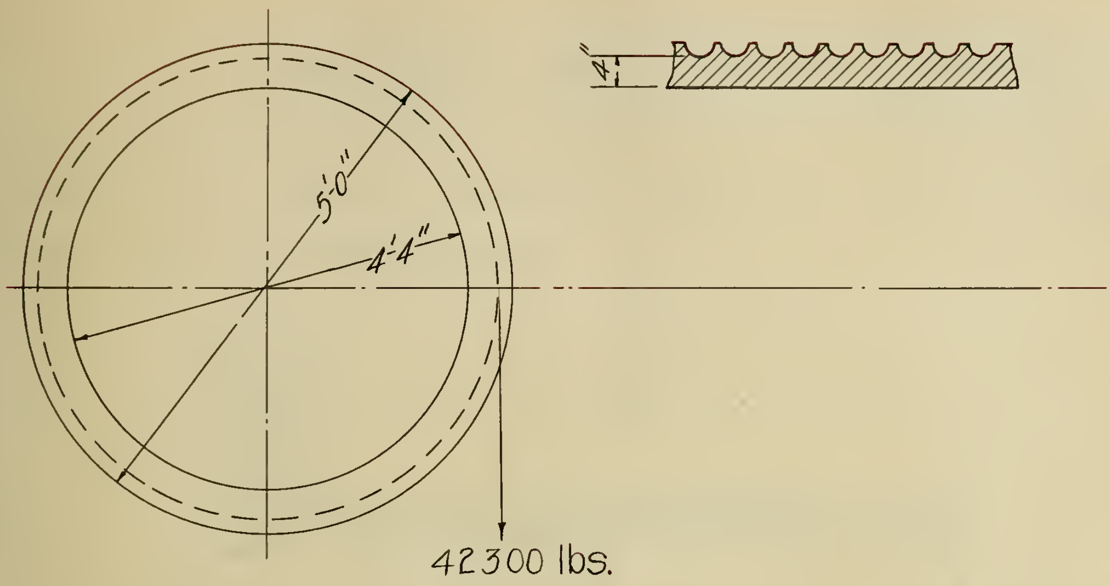


Fig. 16.

$$M_e = \sqrt{M_B{}^2 + M_T{}^2}$$

$$= \sqrt{1269.0^2 + 890.0^2}$$

$$= 1,550,000 \text{ lb.in.}$$

$$\frac{1,550,000}{S} = .098\left(\frac{d^4 - d_1^4}{d}\right)$$

$$= .098\left(\frac{60^4 - 52^4}{60}\right)$$

$$= 168 \text{ lb. per sq.inch}$$

$$\triangle = (P - \tfrac{5}{8}W)\,\frac{l^3}{48\ EI}$$

$$W = 20000 \qquad I = .049\ (d^4 - d_1{}^4) = 2770$$

$$= (42300 - \tfrac{5}{8}20000)\,\frac{84^3}{48 \times 15000000 \times 2770}$$

$= .0168$ inch   Note: Deflection governs design. The drum will be ribbed in various places to decrease this deflection.

27. Design of Drum Shaft - The drum is fitted with bronze bushings and turns freely upon the stationary shaft thus eliminating the twisting moment on the shaft. Fig. 17 shows the forces which act on the shaft.

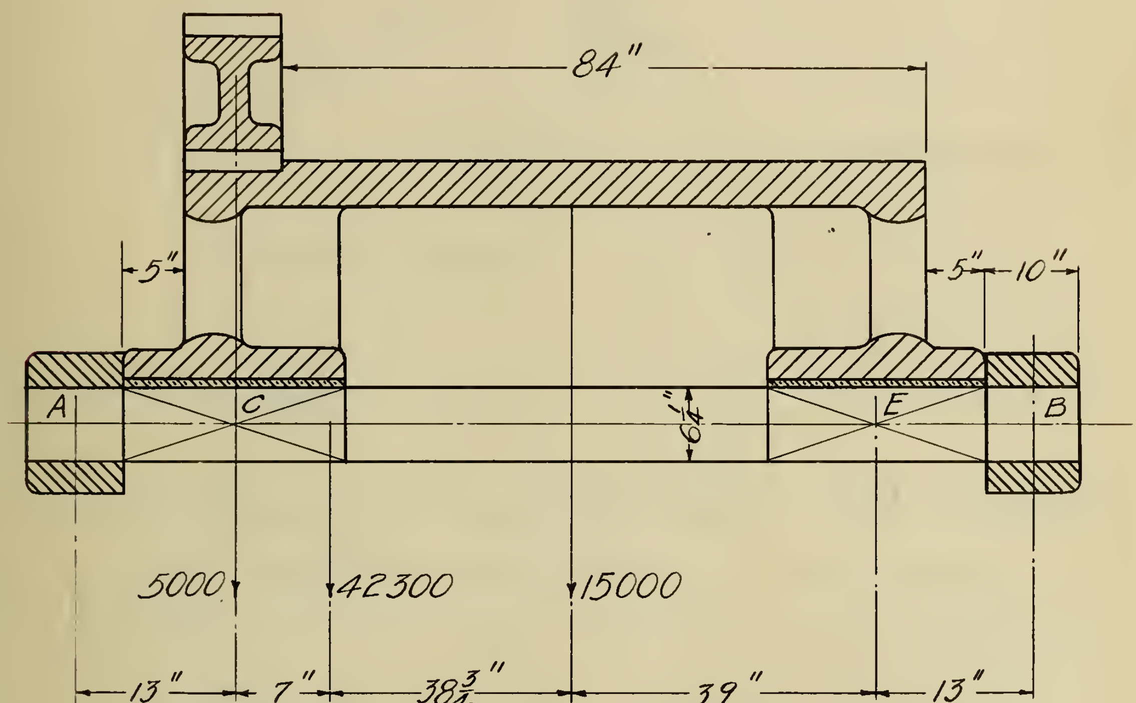


Fig. 17.

Load on tooth acting 30° with horizontal 29400 lb.

$$30^\circ \text{ component at A} = \frac{29400}{110.75}(110.75 - 13)$$

$$= 25950 \text{ lb.}$$

$$V_A = \frac{5000 \times 97.75 + 42300 \times 90.75 + 15000 \times 52}{110.75}$$

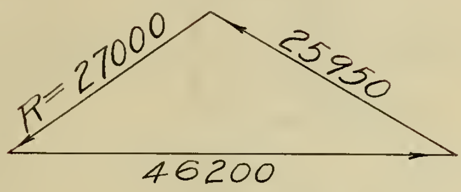
$$= 46200 \text{ lb.}$$

By combining graphically the two forces acting at A as shown in Fig. 18 we have as a resultant A_R= 27000 lb. and the bending moment due to this force

$$M_B = 27000 \times 13 = 351000 \text{ in.lb.}$$

$$\frac{351000}{16000} = \frac{I}{c} = 21.9$$

Diameter of shaft = 4 7/8 inches



Scale: 1inch = 20000 lb.

Fig. 18.

<u>Bearing Pressure</u> -

Projected area = 4 7/8 x 16 = 78 sq.in.

$$\frac{27000}{78} = 346 \text{ lb.}$$

28. <u>Design of Double Block Brake</u> - The differential band brake was originally selected for the case at hand but upon investigation it was found that a band was required which was too thick for satisfactory operation. The double block brake Fig. 19 is used to considerable advantage as far as convenience of application to the trolley is concerned. Knowing the horse-power to be taken care of, the resisting force

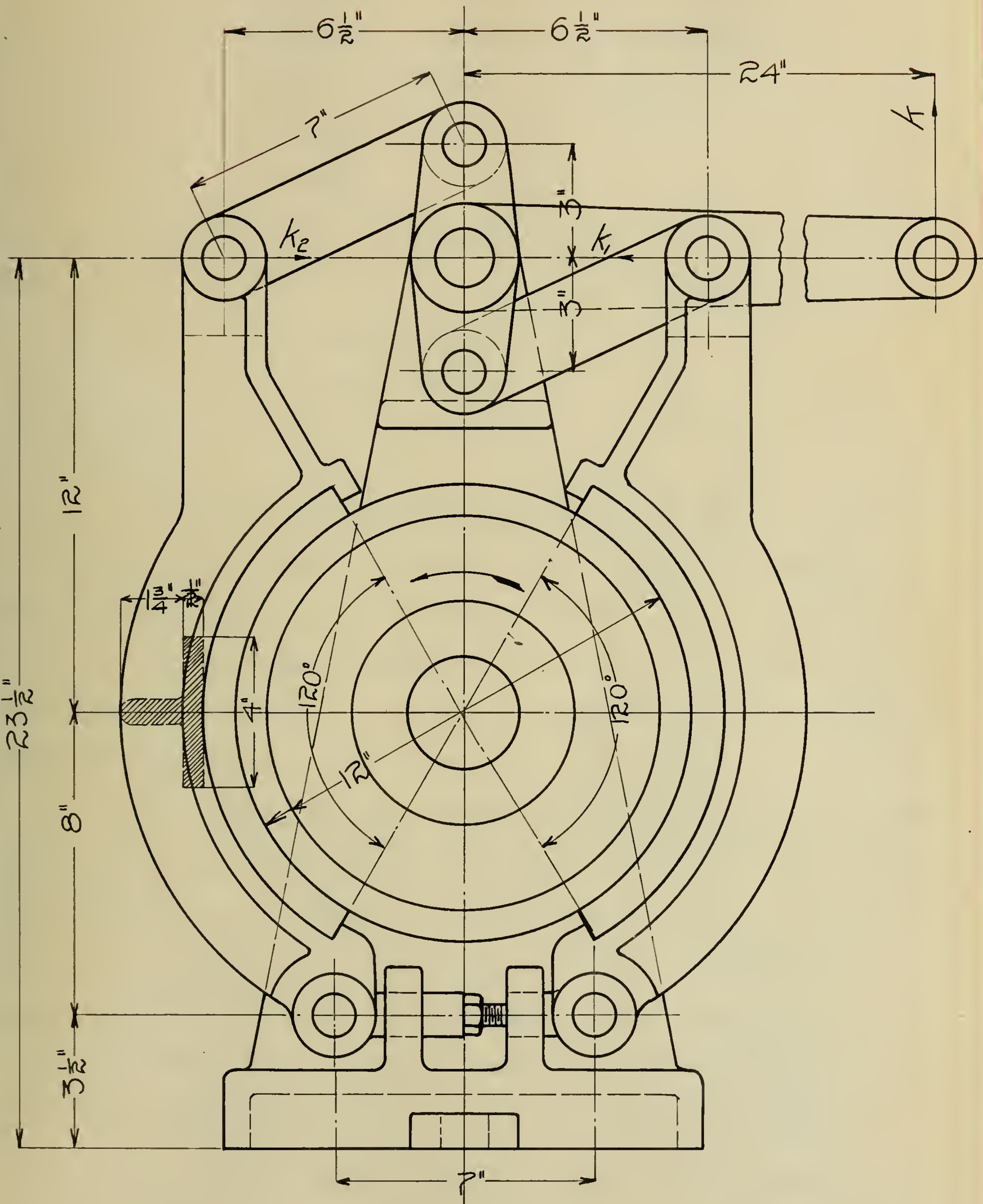
$$F = \frac{40 \times 33000 \times 12}{12 \times \quad \times 400} = 1050 \text{ lb.}$$

The force on each block is 525 lb. Assuming $\mu = 0.4$ and $2\theta = 120^\circ$ and substituting in the following expression for T we have

$$T = 2\mu P \frac{\sin\theta}{\theta + \sin\theta\cos\theta} = 2 \times .4P \times 0.585$$

from which

$$P = \frac{525}{.468} = 1120 \text{ lb.}$$



Double block brake

Fig. 19.

Scale $\frac{1}{4}$" = 1'

By taking moments about the pin joint k_1 becomes

$$k_1 20 + 525 \times 2.5 - 8 \times 1120 = 0$$

$$k_1 = 383$$

$$k_2 = 514$$

By taking moments about the fulcrum of the lever for the value of k we have

$$k\,24 = 3(k_1 + k_2)$$

$$= 3 \times 897$$

$$k = 112$$

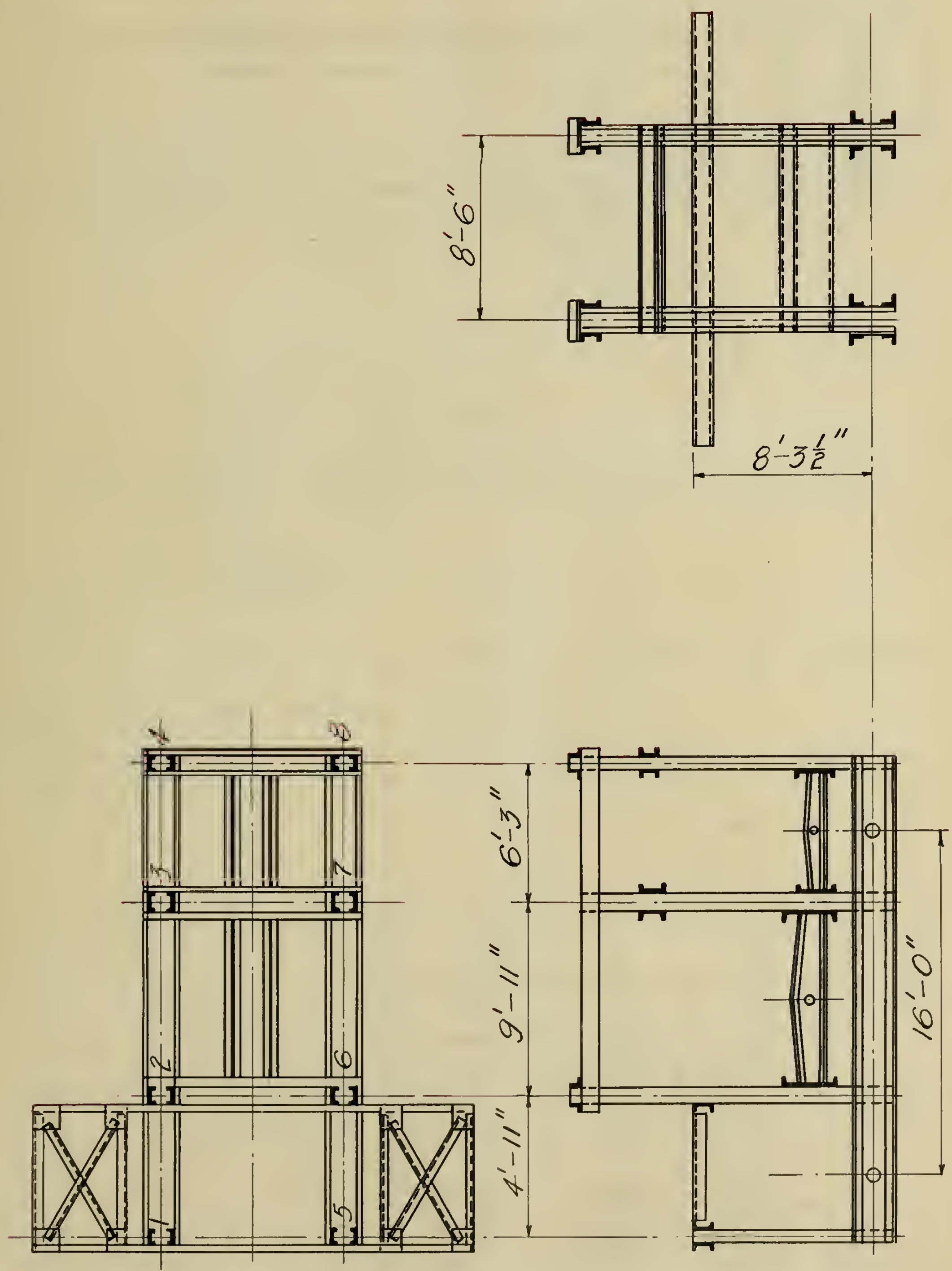
29. Design of Trolley Frame - According to specifications the trolley frame is to be of the rigid frame steel type. Fig. 20 shows such a type of trolley frame. It is composed of riveted structural steel members, the sections of which are determined directly from the forces to which they are subjected. The following gives a rigid design of each member.

30. Design of Main Girder of Trolley Frame - An exact determination of the forces which act on the main girder of the trolley frame would involve laborious mathematical calculations hence for all practical purposes certain assumptions were made in regard to the distribution of the load coming upon the eight columns and therefore upon the girders. The forces shown in Fig.21 are those finally arrived at. The reactions are found to be

$$R_1 = \frac{3.3 \times 92950 + 13.25 \times 92950 - 3 \times 3750 + 2600 \times 18.86}{16}$$

$$= 98700 \text{ lb.}$$

R_2 = 93550 lb. from which the maximum bending moment is



Trolley Frame
Fig. 20.

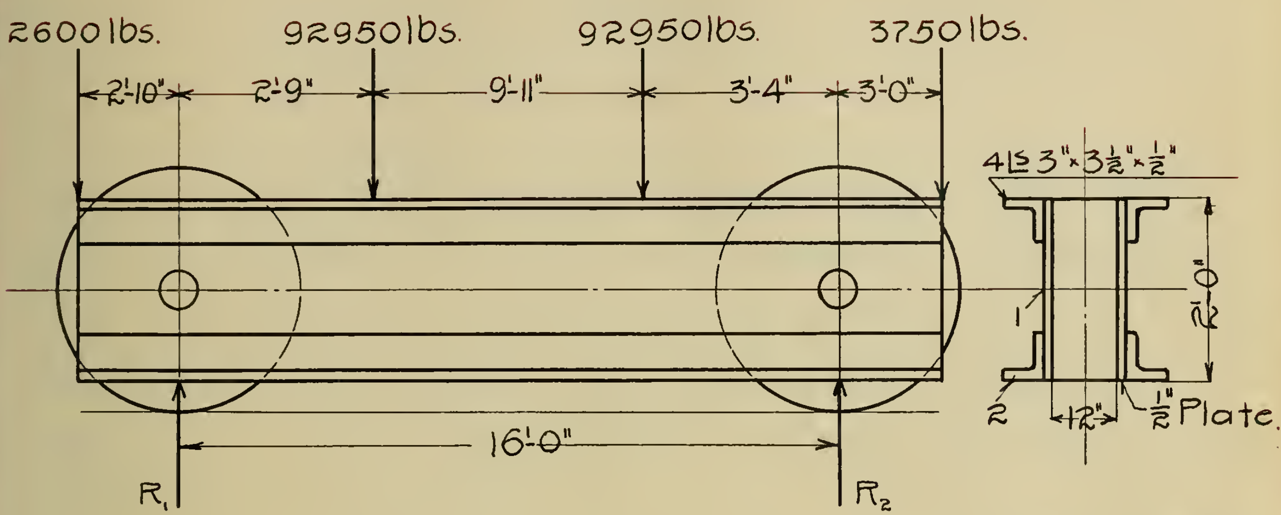


Fig. 21,

6.25 x 3750 - 3.25 x 93550 = 280,580 pound-feet

$$\frac{M}{S} = \frac{I}{c} = \frac{280580 \times 12}{16000} = 210$$

The following table gives the properties of the section assumed and shows the section to be sufficient.

No.	Area in sq.in.	h	I	Ah^2	$\bar{I}$	$\frac{I}{c}$
1	12	0	576	0	576	216.5
1	12	0	576	0	576	
2	3	10.87	3.45	355	358.5	
2	3	10.87	3.45	355	358.5	
2	3	10.87	3.45	355	358.5	
2	3	10.87	3.45	355	358.5	

Maximum shear is 98700 lb.

$$\text{Required area} = \frac{98700}{10000} = 9.87 \text{ sq.in.}$$

Actual area in shear is 10.1 sq.in.

31. Design of Trolley Frame Columns - By a careful analysis of the forces acting on the columns it becomes evident that the horizontal thrust on the drum bearings produces a bending moment in the columns. The complex cross framing does not permit a thorough rational design hence the columns will be designed only for direct stress.

Columns No. 1 and 5

Load on each column 2600 lb.

Length = 7 ft. 2 1/2 in.

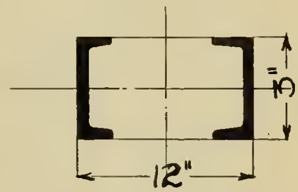


Fig. 22.

A rational design is unnecessary due to the very small load, then minimum size channels will be used and these will be in excess of the area required by the load. Use 2-3in.-4 lb. channels. Tie channels together at appropriate places with 3/8" batten plates. Fig. 22 shows the arrangement of the section.

Columns 2, 3, 6 and 7.

Maximum load 52200 lb.

Length = 7 ft.

Least l/r = 125

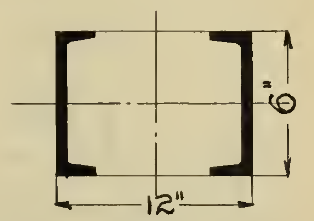


Fig. 23.

Try 2 - 6" - 8.0 lb. channels. Area = 4.7 sq.in.

Least r = 2.33 l/r = 36

$S = 16000 - 70\ l/r$

$= 13480$ lb. per sq.in.

$$\text{Required area} = \frac{52200}{13480} = \text{say } 4 \text{ sq.in.}$$

Use 2 - 6" - 8.0 lb. channels

Fig. 23 shows the arrangement of the section.

Columns No. 4and 8

Load 3750 lb.

Length 7 ft.

Try 2 - 3" - 4 lb. channels

Area = 2.38 sq.in.

Least r = 1.17 $l/r = 71.9$

$S = 16000 - 70\ l/r$

$= 10967$

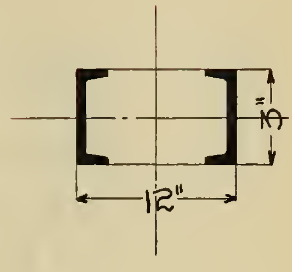


Fig. 24.

$$\text{Required area} = \frac{3750}{10967} = .34 \text{ sq.in.}$$

Use 2 - 3" - 4 lb. channels as shown in sketch.

For reasons of construction use 2 - 6" - 8 lb. channels. Channels are placed as shown in Fig. 24.

32. <u>Design of Trolley Motor Platform</u> - The motor presents the simple case of the cantilever beam Fig. 25 with a maximum bending moment

$$= 1900 \times 48 = 91200 \text{ lb.in.}$$

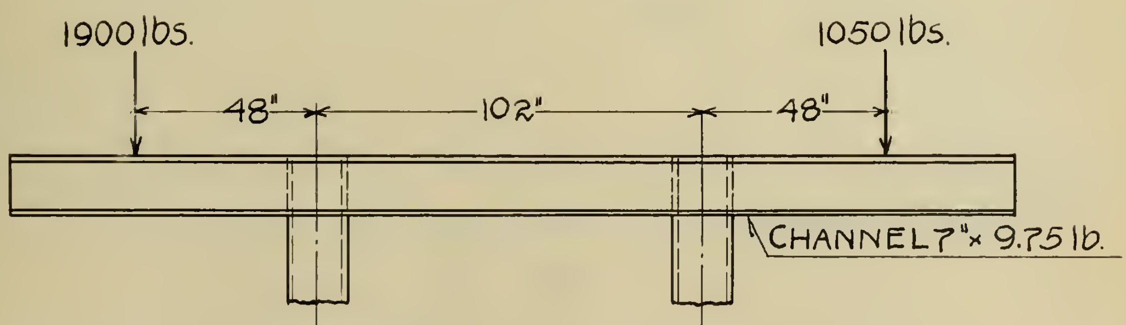


Fig. 25.

$S = 16000$

$$\frac{91200}{16000} = 5.7 = \frac{I}{c}$$

Use a 7" - 9 3/4 lb. channel

33. Design of Cross Beam for Support of Worm Gear Shaft

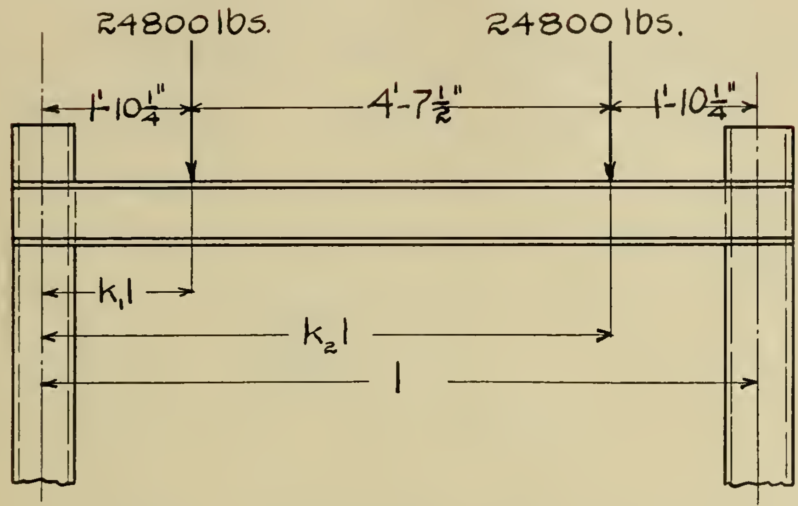


Fig. 26.

The beam will be treated according to the theory given for fixed beams, the condition of loading is given by Fig. 26. The bending moment under the first load is

$$M = +Plk^2(2 - 4k + 2k^2)$$

$$k_1 = \frac{22.25}{102} = 0.218$$

$$M = + 24800 \times 102 \times \overline{.218}^2(2 - 4 \times .218 + 2 \times .218^2)$$

$$= 146500 \text{ lb.in.}$$

Moment under the first load due to the second load

Reaction at left end due to second load is

$$R_1 = P(1 - 3k^2 + 2k^3)$$

$$k = \frac{77.75}{102} = 0.76$$

$$R_1 = 24800\ (1 - 3 \times \overline{.76}^2 + 2 \times \overline{.76}^3)$$

$$= + 3600 \text{ lb.}$$

Moment at the left end is $-Plk(1 - 2k + k^2)$

$$M = -24800 \times 102 \times .76(1 - 2 \times .76 + \overline{.76}^2)$$

$$= -110,000 \text{ lb.in.}$$

The moment under the first load due to the second load is: $+3600 \times 22.25 - 110000 = -29000$ lb.in.

Maximum moment is

$$+ 146500 - 29000 = + 117500 \text{ lb.in.}$$

$$\frac{M}{S} = \frac{I}{c} = \frac{117500}{16000} = 7.35$$

Use 2 - 5" - 9 lb. channels

34. Design of Supporting Beams for Trolley Block- Beam for 150-Ton Load.- The plate girders shown in Figs. 27and 28 will be used to support the trolley block.

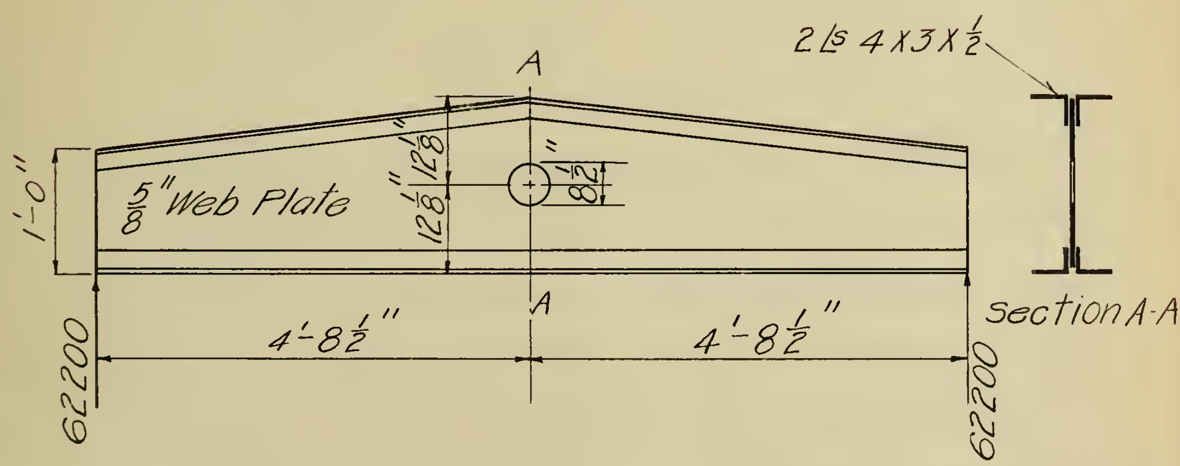


Fig. 27.

$$M = 62200 \times 56\ 1/2$$

$$= 3,510,000 \text{ lb.in.}$$

$$\frac{3,510,000}{16000} = 219.5 = \frac{I}{c}$$

Value of $\frac{I}{c}$ = 222 for the section shown above

Area required in shear at the pin is $\frac{124400}{10000} = 12.44$ sq.in.

Actual area in shear is 23.6 sq.in.

Area required to take end shear is $\frac{124400}{10000} = 12.4$ sq.in.

Actual area in shear is 17.0 sq.in.

Required bearing area of pin $= \frac{124400}{20000} = 6.22$ sq.in.

Actual area $= 5.30$ sq.in.

Rivet a 1/2 in. plate to the side of the girder around the pin hole.

Beam for 60-Ton Load

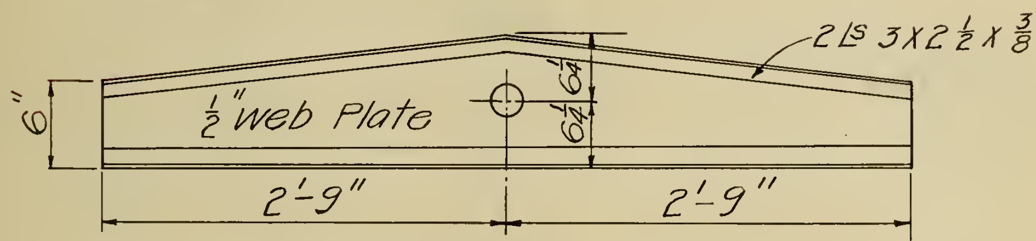


Fig. 28.

35. Design of the Racking Mechanism - Before the design of the racking mechanism can be made it is necessary to determine the total frictional resistance due to the load imposed upon the trolley wheels. The size of trolley wheels according to Ernst is given as follows: Total load on the wheels is 200 tons. The trolley shown diagrammatically in Fig. 29 has 8 wheels, each wheel taking a load of 25 tons. The width of rail is given by the following equation:

$$b = \frac{Q_1}{CD}$$

Q_1 = load per wheel

C = constant = 540

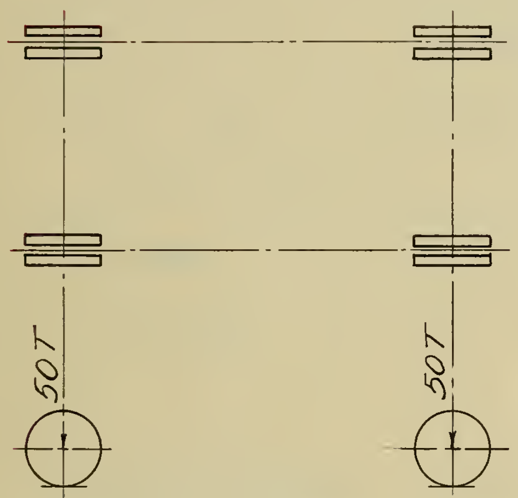
D = diameter of wheel

Assuming a diameter of 36 inches we have as a width of rail

$$b = \frac{50000}{540 \times 36}$$

$$= 2.57 \text{ inches} \quad \text{say } 2\ 9/16 \text{ inches}$$

The wheel proportions are given in Fig. 30.



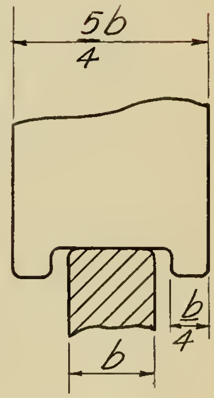


Fig. 30.

Fig. 29.

Weight of 4 standard rails 100 ft. long = 1132 lb.

Knowing the size of trolley wheels and the loads, the resistance due to rolling friction is obtained from the equation:

$$F = c\frac{P}{r}$$

where c = constant for rolling friction (.02)

r = radius of wheel

$$F = .02\ \frac{50000}{18}$$

$$= 555 \text{ lb.}$$

Total = 555 x 8

= 4440 lb.

The flange friction is taken as 2% of total load

$400000 \times .02 = 8000$ lb.

Force required to Overcome Journal Friction

Diameter of journal 4 5/8 in.

$\mu = .1 \quad R = 2\ 5/16$

$M = 2\ 5/16 \times .1 \times 50000 = 11550$ in.lb.

Total moment $= 8 \times 11550 = 92400$ in.lb.

Force at radius $r = \frac{92400}{18} = 5135$ lb.

Total Force to overcome Friction

Journal Friction - - - - -	5135	pounds
Flange " - - - - -	8000	"
Rolling " - - - - -	4440	"
	17575	"

36. Size of Motor - Since the total force required to overcome friction and the speed of racking are known a commercial size of motor is selected as follows:

Efficiency of each pair of spur gears = 96%

" " motor = 80%

Speed of racking assumed as 25 ft. per min.

Combined efficiency $= .96 \times .96 \times .96 \times .80 \times .96 \times$
$\times .96 = .66$

$$H.P. = \frac{17575 \times 12}{.66 \times 33000} = 9.5 \quad \text{say 10 H.P.}$$

Use commercial size 10 H.P. at 400 R.P.M.

$$\frac{1}{G} \times 3 \times 3.14 \times 400 = 25$$

$$G = \frac{3 \times 3.14 \times 400}{25} = 150.6$$

Velocity $= 400 \times 3 \times 3.14 \times \frac{1}{5} \times \frac{1}{5} \times \frac{1}{6} = 25.1$ ft/min.

37. Design of Gears for Racking Mechanism - The arrangement of gears is shown in Fig. 31.

Pinion "B"

$$\text{Load } W = \frac{17575 \times 18}{18.75} = 16900 \text{ lb.}$$

Force on one wheel at radius 18.75" = 8450 lb.

$$\text{Velocity} = \frac{\times 7.5 \times 13.33}{12} = 26 \text{ feet per minute}$$

For trial substitution in the Lewis' formula assume the following dimensions:

$p' = 1.571$; $p = 2$; $f = 3.75$; $n = 15$; $D = 7.5$;

$y = 0.074$ and S for steel casting has the value 20000

$$8450 = 20000 \times 3.75 \times 1.571 \times .074$$

$$\neq 8725$$

The agreement above is close enough , hence the assumed dimensions will be chosen. The diameter of the gear is 37 1/2 inches. An investigation for the strength of the gear is not necessary since the gear and pinion are made of the same material and the pinion is always the weaker.

Pinion "D"

$$\text{Load } W = \frac{8450 \times 3.75}{15} = 2110 \text{ lb.}$$

$$\text{Velocity} = \frac{3.14 \times 6 \times 66.66}{12} = 104 \text{ feet per minute}$$

Assuming the following values for a trial substitution in the Lewis' formula

$p = 2.5$; $p' = 1.257$; $f = 3.5$; $y = 0.068$; $D = 6$;

$n = 15$; and S for cast iron has the value 7000

$$2110 = 7000 \times .068 \times 3.5 \times 1.257 = 2090 \text{ lb.}$$

The above agreement is sufficiently close. The gear "C" does not require investigation for the same reasons given in the previous paragraph.

Pinion "F"

$$\text{Load } W = 2\ \frac{2110 \times 3}{15} = 844 \text{ lb.}$$

$$\text{Velocity} = \frac{3.14 \times 5 \times 400}{12} = 523 \text{ feet per minute}$$

Substituting the following trial values in the Lewis' formula:

$p = 3$; $p' = 1.047$; $f = 2\ 3/4$; $n = 15$; $D = 5$;

$y = .067$ and for cast iron $S = 4500$

$$844 = 4500 \times .067 \times 2.75 \times 1.047$$

$$\neq 870 \text{ lb.}$$

The agreement shows that the trial dimensions may be used and we have 5 inches for the pinion diameter and 30 inches for the gear. For the same reasons given in previous paragraph concerning the strength of gears, gear "E" requires no investigation.

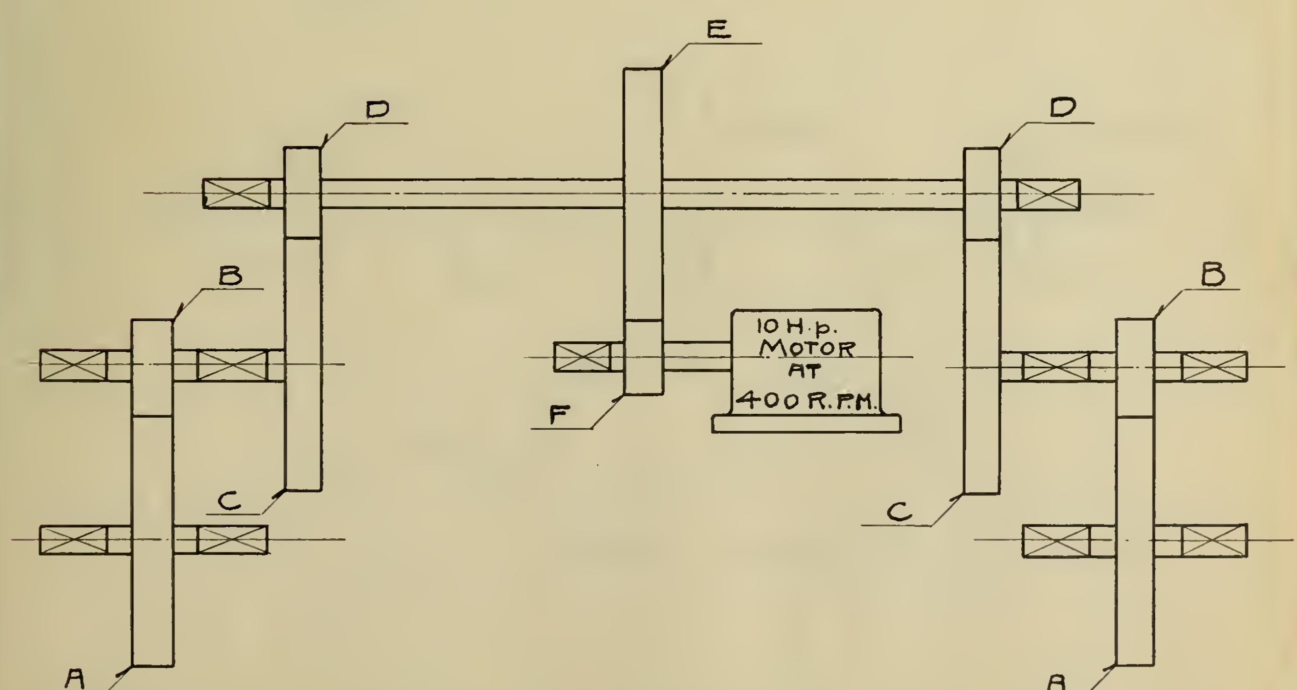


Fig. 31.

38. Design of Shafts for Racking Mechanism - Figs. 32 and 33 shows the arrangement of gears on the shafts and the direction and magnitude of the forces acting upon them.

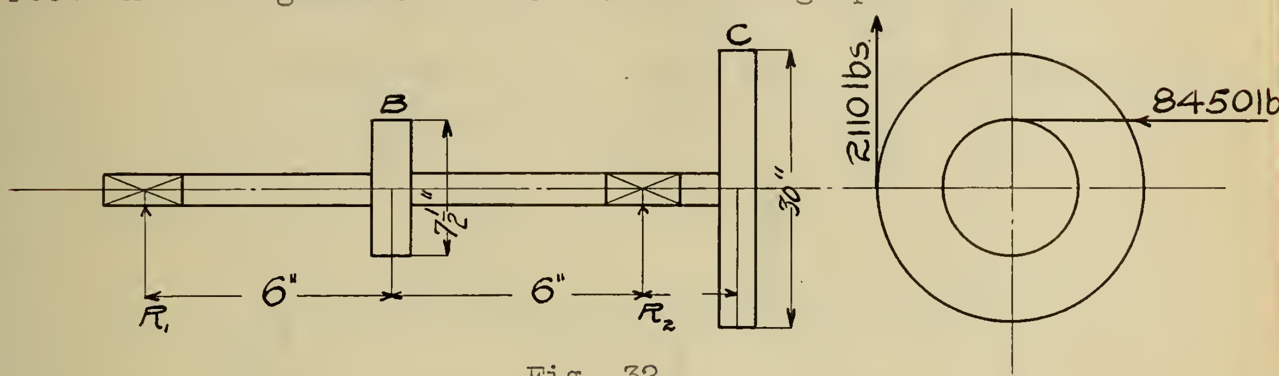


Fig. 32.

$$R_1' = \frac{6 \times 8450}{12} = 4225 \text{ lb.}$$

The maximum bending moment occurs under the small wheel and we have

$$4225 \times 6 = 25350 \text{ lb.in.}$$

$$T = 2110 \times 15 = 31,650 \text{ lb.in.}$$

$$M_e = \sqrt{\overline{31650}^2 + \overline{25350}^2} = 40600 \text{ lb.in.}$$

$$\frac{40600}{16000} = 2.54 = 2\ 15/16 \text{ in. say } 3 \text{ in.}$$

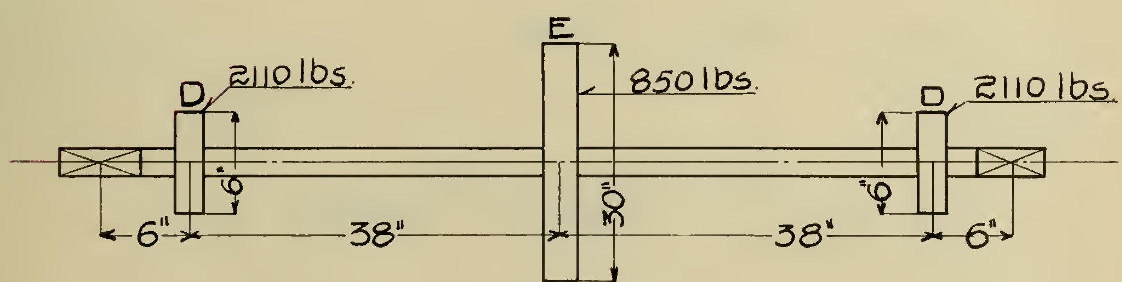


Fig. 33.

$$R_1 = \frac{850 \times 44}{88} = 425 \text{ lb.}$$

We have as a maximum bending moment under the gear

$$425 \times 44 = 18700 \text{ in.lb.}$$

$$T = 850 \times 15 \times 12750 \text{ in.lb.}$$

$$M_e = \sqrt{\overline{18700}^2 + \overline{12750}^2} = 22700 \text{ in.lb.}$$

$$\frac{22700}{16000} = 1.42 = 2\ 7/16 \text{ in. diameter of shaft}$$

39. <u>Design of Trolley Wheel Axle</u> - Fig. 34 shows the forces acting on the axle.

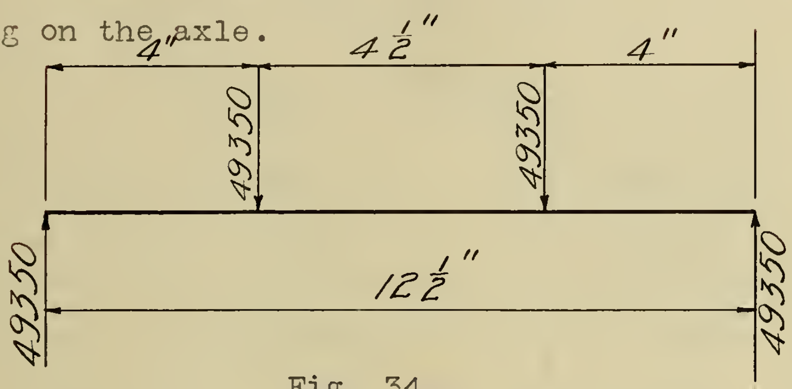


Fig. 34.

The maximum bending moment is

$$49350 \times 4 = 197,500 \text{ lb.in.}$$

Maximum bending moment due to racking mechanism is

$$M_R = \frac{8780 \times 12.5}{4} = 27450 \text{ lb.in.}$$

Maximum combined moment is

$$M = \sqrt{\overline{197500}^2 + \overline{27450}^2} = 199300 \text{ lb.in.}$$

Twisting moment is

$$8780 \times 15 = 131,700 \text{ lb.in.}$$

$$M_e = \sqrt{\overline{131700}^2 + \overline{199300}^2} = 242,000 \text{ lb.in.}$$

$$\frac{M_e}{S} = \frac{242000}{16000} = 15.1 \text{ in}^3.$$

Diameter of axles is 5 3/8 in.

Allowable unit bearing pressure 20000 lb.

Total bearing pressure 49,350 lb.

Required area is $\frac{49350}{20000} = 2.47$ sq.in.

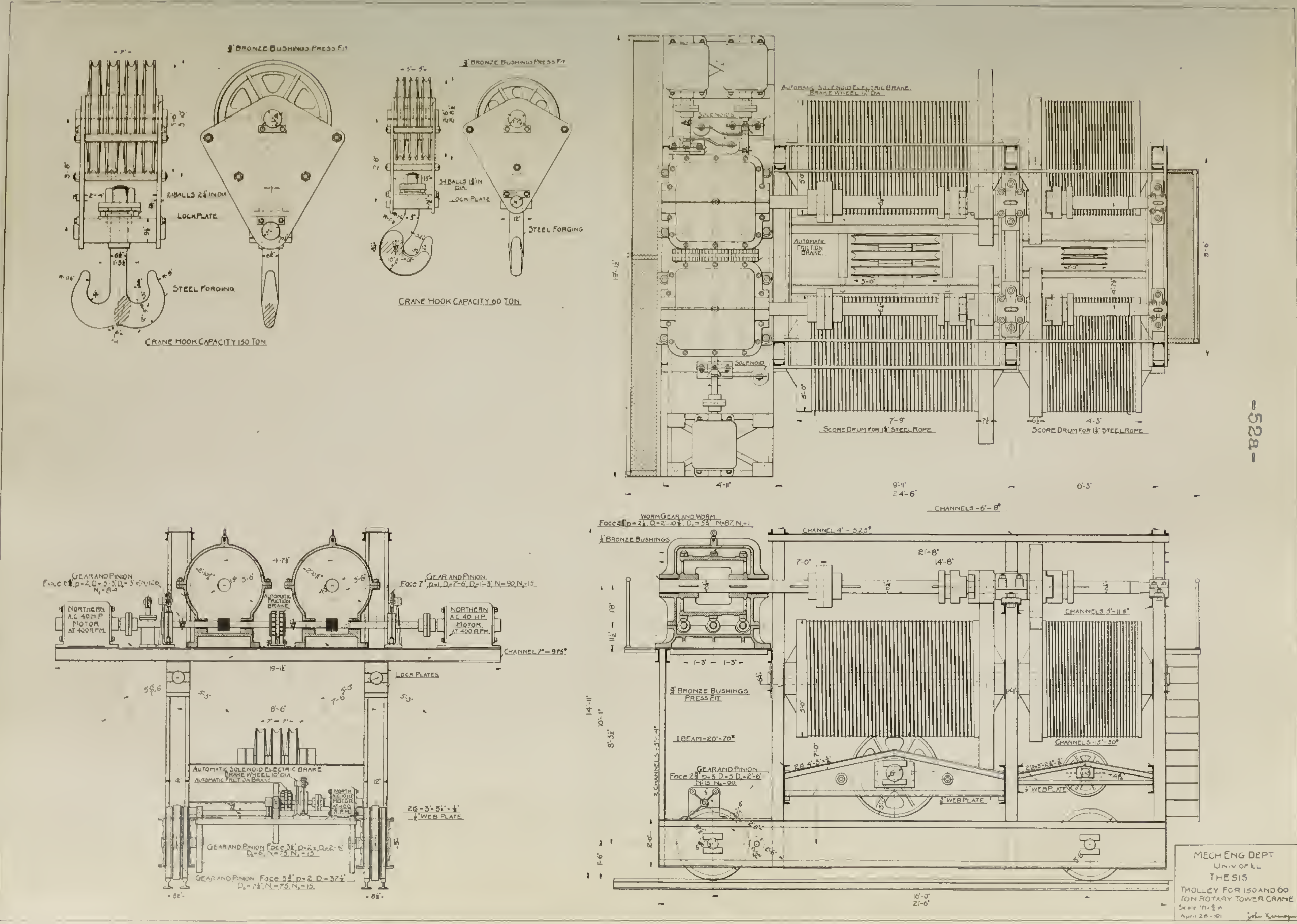
Actual area = 20 sq.in.

40. Weight of Trolley - The weight of the trolley can now be determined and is given in tabular form in the following table.

Table I.

Weight of Trolley

No. of Parts	Name of Parts	Weight in Pounds
2	Block	16800
2	Rope	8975
2	Trolley Block	600
4	Drum	45000
4	Pinion	200
5	Shaft	4240
6	Worm Gears	2200
1	Worm Gear Case	1100
3	40 H.P. Motor	5100
1	10 H.P. Motor	450
1	Trolley Frame	6400
8	Trolley Wheel	7000
1	Racking Mechanism	400
	Total	98465



CRANE HOOK CAPACITY 150 TON
CRANE HOOK CAPACITY 60 TON
STEEL FORGING
LOCK PLATE
BRONZE BUSHINGS PRESS FIT
34 BALLS 1½" IN DIA.
AUTOMATIC SOLENOID ELECTRIC BRAKE
AUTOMATIC FRICTION BRAKE
SOLENOID
SCORE DRUM FOR 1¼" STEEL ROPE
CHANNELS - 6'- 8"
WORM GEAR AND WORM
BRONZE BUSHINGS
CHANNEL 4" - 5.25"
CHANNELS 5" - 11.5"
BRONZE BUSHINGS PRESS FIT
I BEAM - 20" - 70"
GEAR AND PINION
2 CHANNELS
CHANNELS - 15" - 50"
WEB PLATE
NORTHERN A.C. 40 H.P. MOTOR AT 400 R.P.M.
CHANNEL 7" - 9.75"
LOCK PLATES
AUTOMATIC SOLENOID ELECTRIC BRAKE BRAKE WHEEL 10" DIA
AUTOMATIC FRICTION BRAKE
MECH ENG DEPT
UNIV OF ILL
THESIS
TROLLEY FOR 150 AND 60 TON ROTARY TOWER CRANE
April 28 - 1911

Chapter V

The Structure

41. Theory of the Structure - Knowing the weight of the trolley and the maximum load to be lifted sufficient data is now at hand for determining the counterweight required to balance the structure. Fig. 35 gives a diagram of structure and the forces acting upon it.

A general equation for the maximum horizontal reaction H will be derived assuming that the overturning moment, when the crane is loaded, is equal to the overturning moment with the crane empty.

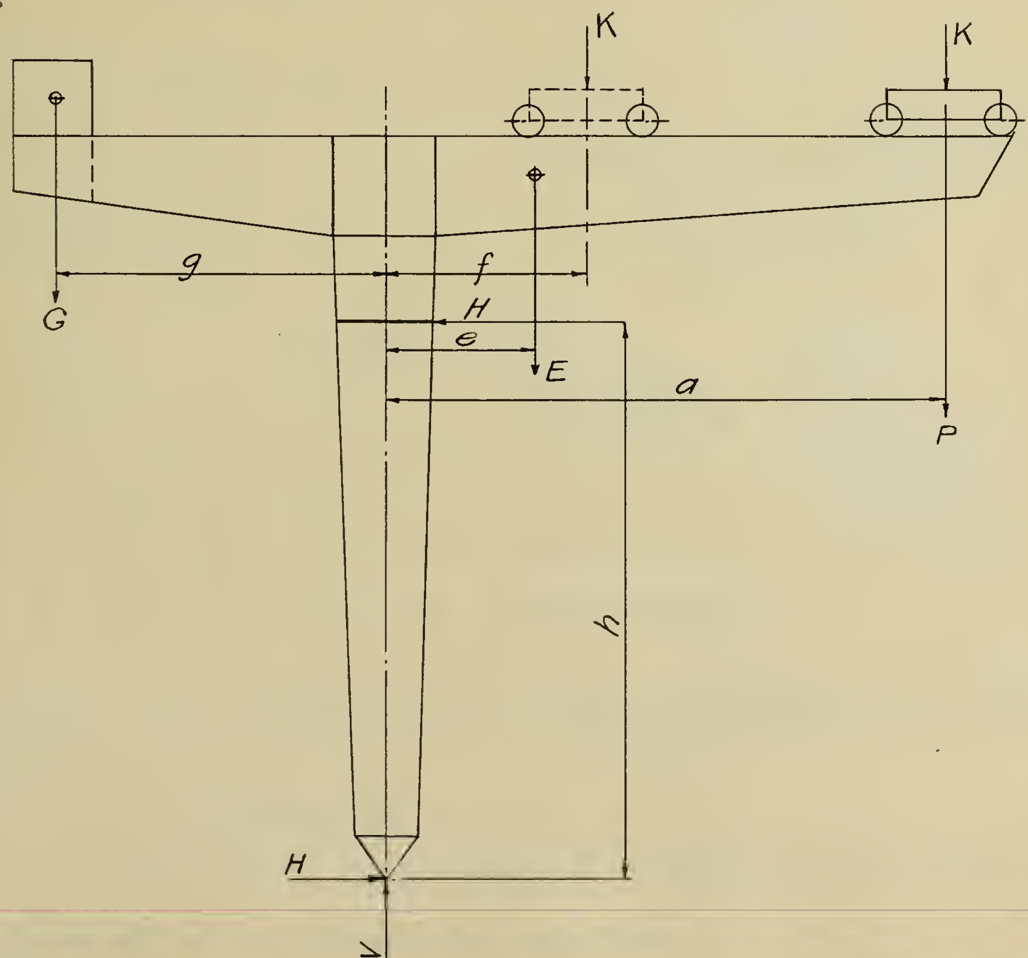


Fig. 35.

Let M_l = overturning moment(crane loaded)

M_u = " " (crane unloaded)

and taking moments about the pivot we have:

$$M_l = P_a + K_a + E_e - G_g \text{ - - - - - - - - - - - - } (8)$$

$$M_u = -Kf + G_g - E_e \text{ - - - - - - - - - - - - - - } (9)$$

Solve for G:

$$G = \frac{2E_e + P_a + K_a + K_f}{2\ g} \text{ - - - - - - - - - - - - } (10)$$

Substitute (10) in (9)

$$M_{max} = \frac{P}{2}a + \frac{K}{2}(a - f) - \text{ - - - - - - - - - - - - } (11)$$

Divide thru by k we have

$$H_{max} = \frac{1}{h}\left[\frac{P}{2}a + \frac{K}{2}(a - f)\right] \text{ - - - - - - - - - - } (12)$$

Assume E = 220 tons, weight of revolving structure, one half of which or 110 tons is taken by each truss.

Load on each lower panel point

$$\frac{220,000}{11} = 20,000 \text{ lb.}$$

End panel points take 10000 lb.

Center of gravity of all the forces is 20 ft. to the right of the center line of the central tower. Substituting in equation (10) we have for the counter-weight

$$G = \frac{2 \times 220 \times 20 + 150 \times 112 + 50 \times 112 + 50 \times 28}{2 \times 77.5}$$

$$= 210 \text{ tons}$$

Half of this counterweight or 105 tons is taken by each truss from which the horizontal reaction

$$H = \frac{1}{120}\left[\frac{75 \times 112}{2} + \frac{25}{2}(112 - 28)\right]$$

$$= 43.75 \text{ tons}$$

The vertical reaction

$$V = 210 + 150 + 50 + 220$$

$$= 630 \text{ tons}$$

With the horizontal and vertical reactions known the stresses in the members due to the dead and live loads are determined according to the principles of graphic statics. Figs. 36 and 37 show the stress diagrams for the conditions of loading which produce maximum stresses.

42. Stability of the Structure - In order to determine whether or not the crane is safe against overturning due to the external forces the following tables of stability give the conditions of loading and the method of finding the distance out from the center of the structure at which the center of gravity falls. The conditions given in table VI are the most unfavorable, indicating that the base of the outer tower must be at least 60 feet square and consequently this dimension was chosen.

Tables of Stability

Table II

Superstructure only with 150 Ton Load

	Tons	Feet	Moments +	Moments -
Working Load	150	110	16500	
Trolley and Block	50	110	5500	
Revolving Jib	220	20	4400	
Counterbalance	210	77		16200
	630		26400	16200

$$\frac{26400 - 16200}{630} = 16.2 \text{ feet from the center}$$

Table III

Superstructure only with no Load

	Tons	Feet	Moments +	Moments -
No Load	0	0	0	
Trolley and Block	50	28	1400	
Revolving Jib	220	20	4400	
Counterbalance	210	77		16200
	480		5800	16200

$$\frac{16200 - 5800}{480} = 21.7 \text{ feet}$$

Table IV

Crane Complete with 150 Ton Load

	Tons	Feet	Moments +	Moments -
Superstructure and Load	630	16.2	10200	
Tower with Roller Path	200	0.0		0
	830		10200	

$$\frac{10200}{830} = 12.3 \text{ feet from center}$$

Table V

Crane Complete with No Load

	Tons	Feet	Moments +	Moments -
Superstructure & No Load	480	21.7	10400	
Tower with Roller Path	200	0.0		0
	680		10400	

$$\frac{10400}{680} = 15.3 \text{ feet from center}$$

Wind Forces

When the crane is unloaded the wind force produces its greatest effect. A lateral force of 250 pounds per lineal foot of upper and lower chord of truss will be assumed acting at the center line of the truss, and a lateral force of 100 pounds for each vertical lineal foot of the tower. (Cooper's Bridge Specifications)

Total wind force on truss

2(210 ft. x 250) = 105000 lb. at 160 ft. above ground

Revolving jib columns and bracing

140 ft. x 100 = 14000 lb. at 70 ft. above ground

Tower columns and bracing

140 x 100 = 14000 lb. at 62 ft. above ground

Moment of wind forces

$$\frac{105000 \times 160 - 14000 \times 70 - 14000 \times 62}{105000 - 14000 - 14000} = 140 \text{ feet above ground}$$

Overturning effect of wind

Table VI

Crane with no Load and Maximum Wind Force Acting

	Tons	Feet	Moments +	Moments -
Crane Complete	680	15.3	10400	0
Wind	66.5	140.0	9300	0
	680		19700	

$$\frac{19700}{680} = 29 \text{ feet from center}$$

Discussion: The calculations show that with the crane loaded with the maximum load at 110 feet radius the center of gravity of the superstructure is 16.2 feet from the center of rotation, and with the crane unloaded 21.7 feet. Even though the crane is empty and there is a hurricane blowing the center of gravity of the crane which is 29 feet from the center is never outside the columns of the foot of the tower. Under these circumstances the foundation bolts are never stressed. It is not probable that the most unfavorable condition as represented un-

der Table VI will ever occur.

43. Stresses in the Rotary Jib - A graphical solution of the stresses in the members of the jib is given in Figs. 36 and 37. The loadings for which the stresses were determined are shown on the stress sheet. The following table gives the stresses due to the various loadings together with the maximum and minimum stresses.

Table VII

Member	Trolley at outer position with maximum load	Trolley at inner position without load	Maximum Stress lb.	Minimum Stress lb.
	Dead load + live load stresses lb.	Dead load + load stresses due to wt. of trolley- lb.		
X-1	-380,000	-398,000	-398,000	-380,000
X-2	-268,000	-390,000	-390,000	-268,000
X-4	-126,000	-480,000	-480,000	-126,000
X-6	- 60,000	-556,000	-556,000	- 60,000
X-8	+ 10,000	-620,000	-620,000	+ 10,000
X-10	+ 80,000	-674,000	-674,000	+ 80,000
X-12	+134,000	-722,000	-722,000	+134,000
X-14	+180,000	-762,000	-762,000	+180,000
X-16	+222,000	-802,000	-802,000	+222,000
X-19	+226,000	-802,000	-802,000	+226,000
X-20	- 40,000	- 40,000	- 40,000	- 40,000
X-21	- 18,000	- 18,000	- 18,000	- 18,000
X-24	-320,000	-320,000	-320,000	-320,000

Table VII cont.

Member	Trolley at outer position with maximum load	Trolley at inner position without load	Maximum Stress lb.	Minimum Stress lb.
	Dead load + live load stresses lb.	Dead load + load stresses due to wt. of trolley- lb.		
X -27	-582,000	-586,000	-586,000	-582,000
X -30	-782,000	-786,000	-786,000	-782,000
X'-1	-508,000	-280,000	-508,000	-280,000
X'-3	-476,000	-200,000	-476,000	-200,000
X'-5	-568,000	- 90,000	-568,000	- 90,000
X'-7	-644,000	0	-644,000	0
X'-9	-710,000	+ 72,000	-710,000	+ 72,000
X'-11	-764,000	+140,000	-764,000	+140,000
X'-13	-812,000	+194,000	-812,000	+194,000
X'-15	-852,000	+240,000	-852,000	+240,000
X'-17	-890,000	+284,000	-890,000	+284,000
X'-18	-890,000	+288,000	-890,000	+288,000
X'-36	-1280,000	-334,000	-1280,000	-334,000
X'-39	-1066,000	-228,000	-1066,000	-228,000
X'-42	-828,000	-152,000	-828,000	-152,000
X'-45	-550,000	- 84,000	-550,000	- 84,000
X'-48	-234,000	- 32,000	-234,000	- 32,000
X'-51	0	- 16,000	- 16,000	0
X'-52	0	0	0	0
Y-20	+ 18,000	+ 16,000	+18,000	+ 16,000
Y-22	+146,000	+146,000	+146,000	+146,000

Table VII cont.

Member	Trolley at outer position with maximum load	Trolley at inner position without load	Maximum Stress lb.	Minimum Stress lb.
	Dead load + live load stresses lb.	Dead load + load stresses due to wt. of trolley- lb.		
Y-23	+146,000	+146,000	+146,000	+146,000
Y-25	+452,000	+454,000	+454,000	+452,000
Y-26	+452,000	+454,000	+454,000	+452,000
Y-28	+680,000	+684,000	+684,000	+680,000
Y-29	+680,000	+684,000	+684,000	+680,000
Y-31	+860,000	+860,000	+860,000	+860,000
Y-32	+860,000	+860,000	+860,000	+860,000
Y-34	+1375,000	+396,000	+1375,000	+396,000
Y-35	+1375,000	+396,000	+1375,000	+396,000
Y-37	+990,000	+274,000	+990,000	+274,000
Y-38	+990,000	+274,000	+990,000	+274,000
Y-40	+944,000	+184,000	+944,000	+184,000
Y-41	+944,000	+184,000	+944,000	+184,000
Y-43	+684,000	+112,000	+684,000	+112,000
Y-44	+684,000	+112,000	+684,000	+112,000
Y-46	+392,000	+ 92,000	+392,000	+ 92,000
Y-47	+392,000	+ 92,000	+392,000	+ 92,000
Y-49	+100,000	+ 8,000	+100,000	+ 8,000
Y-50	+100,000	+ 8,000	+100,000	+ 8,000
Y-52	0	0	0	0

Table VII cont.

Member	Trolley at outer position with maximum load Dead load + live load stresses lb.	Trolley at inner position without load Dead load + load stresses due to wt. of trolley- lb.	Maximum Stress lb.	Minimum Stress lb.
a	+260,000	+198,000	+260,000	+198,000
b	- 76,000	- 76,000	- 76,000	- 76,000
c	- 66,000	- 70,000	- 70,000	- 66,000
d	- 62,000	- 64,000	- 64,000	- 62,000
e	- 57,000	- 60,000	- 60,000	- 57,000
f	- 53,000	- 56,000	- 56,000	- 53,000
g	- 50,000	- 52,000	- 52,000	- 50,000
h	- 46,000	- 48,000	- 48,000	- 46,000
i	- 86,000	- 88,000	- 88,000	- 86,000
j	-1126,000	-640,000	-1126,000	-640,000
2-3	+178,000	+136,000	+178,000	+136,000
4-5	+116,000	+116,000	+116,000	+116,000
6-7	+ 98,000	+102,000	+102,000	+ 98,000
8-9	+ 88,000	+ 90,000	+ 90,000	+ 88,000
10-11	+ 78,000	+ 80,000	+ 80,000	+ 78,000
12-13	+ 70,000	+ 72,000	+ 72,000	+ 70,000
14-15	+ 64,000	+ 66,000	+ 66,000	+ 64,000
16-17	+58,000	+ 60,000	+ 60,000	+ 58,000
18-19	+ 52,000	+ 52,000	+ 52,000	+ 52,000
20-21	- 42,000	- 42,000	- 42,000	- 42,000
21-22	-194,000	-196,000	-196,000	-194,000

Table VII cont.

Member	Trolley at outer position with maximum load	Trolley at inner position without load	Maximum Stress lb.	Minimum Stress lb.
	Dead load + live load stresses lb.	Dead load + load stresses due to wt. of trolley- lb.		
22-23	- 70,000	-140,000	-140,000	- 70,000
23-24	+260,000	+260,000	+260,000	+260,000
24-25	-238,000	-240,000	-240,000	-238,000
25-26	0	0	0	0
26-27	+216,000	+216,000	+216,000	+216,000
27-28	-206,000	-208,000	-208,000	-206,000
28-29	0	0	0	0
29-30	+196,000	+196,000	+196,000	+196,000
30-31	-190,000	-198,000	-198,000	-190,000
31-32	0	0	0	0
32-33	+586,000	-518,000	+586,000	-518,000
33-34	-584,000	+520,000	-584,000	+520,000
34-35	0	0	0	0
35-36	-216,000	-144,000	-216,000	-144,000
36-37	+220,000	+118,000	+220,000	+118,000
37-38	0	0	0	0
38-39	-220,000	- 20,000	-220,000	- 20,000
39-40	+226,000	+ 80,000	+226,000	+ 80,000
40-41	0	0	0	0
41-42	-228,000	- 64,000	-228,000	- 64,000
42-43	+240,000	+ 66,000	+240,000	+ 66,000

Table VII conc.

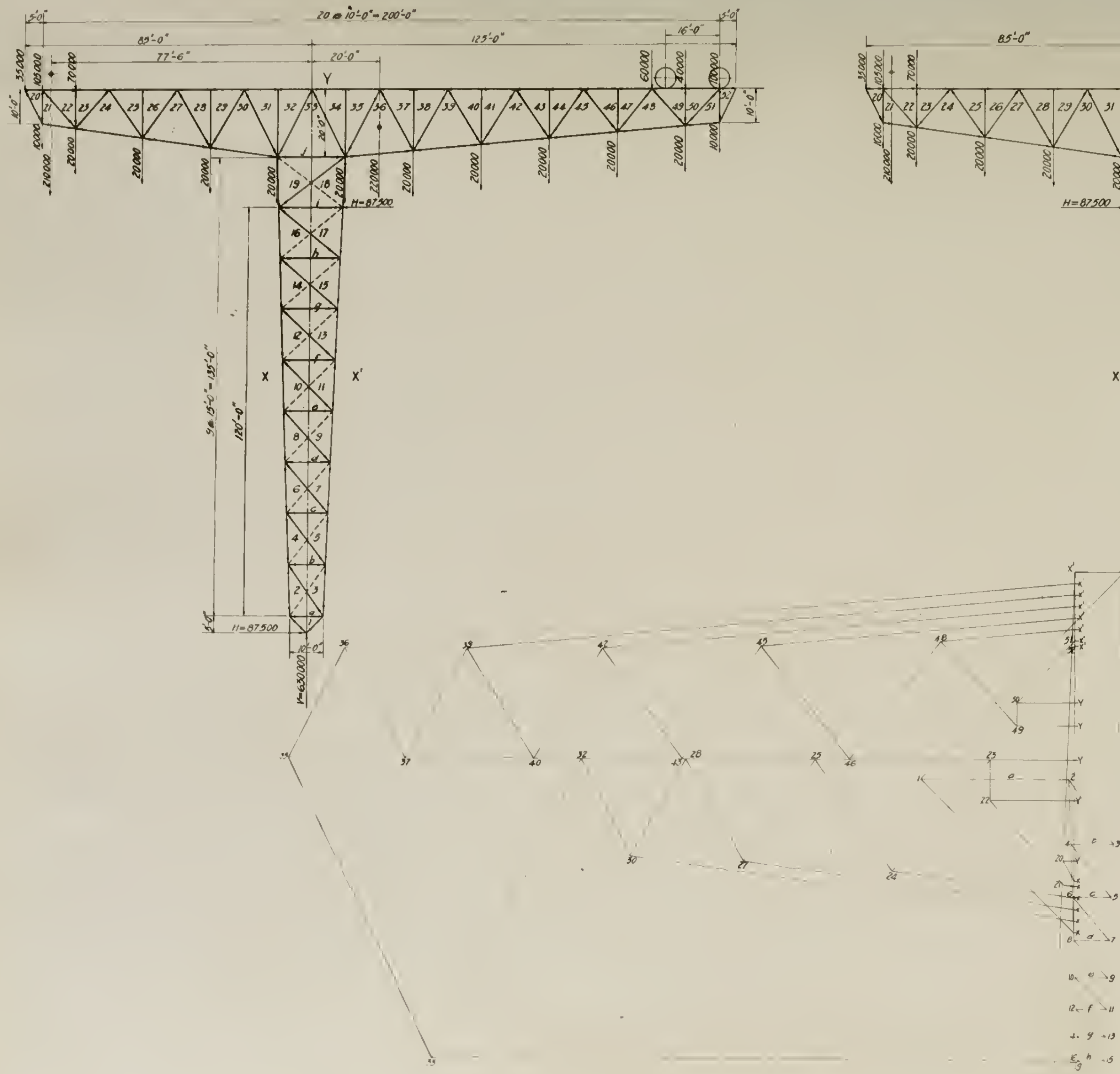
Member	Trolley at outer position with maximum load	Trolley at inner position without load	Maximum Stress	Minimum Stress
	Dead load + live load stresses lb.	Dead load + load stresses due to wt. of trolley- lb.	lb.	lb.
43-44	0	0	0	0
44-45	-240,000	- 50,000	-240,000	- 50,000
45-46	+246,000	+ 52,000	+246,000	+ 52,000
46-47	0	0	0	0
47-48	-260,000	- 34,000	-260,000	-34,000
48-49	+200,000	+ 34,000	+200,000	+ 34,000
49-50	- 80,000	0	- 80,000	0
50-51	-148,000	- 12,000	-148,000	- 12,000
51-52	+ 20,000	+ 20,000	+ 20,000	+ 20,000

44. Tower Stresses - Fig. 38 shows a line drawing of the outside tower and the loads acting on it. Each panel point of the tower is subjected to the wind load acting on half a panel length on either side of the point under consideration. The vertical forces due to the weight of the tower were distributed according to the best judgement of the writers since no rational method is known. In Figs. 39-43 is given a graphical solution of the stresses in the members of the tower due to dead and wind loads, the wind acting in either direction. The stresses in the members of the tower due to the different loadings, together with the maximum and minimum stresses are shown in Table VIII.

Table VIII

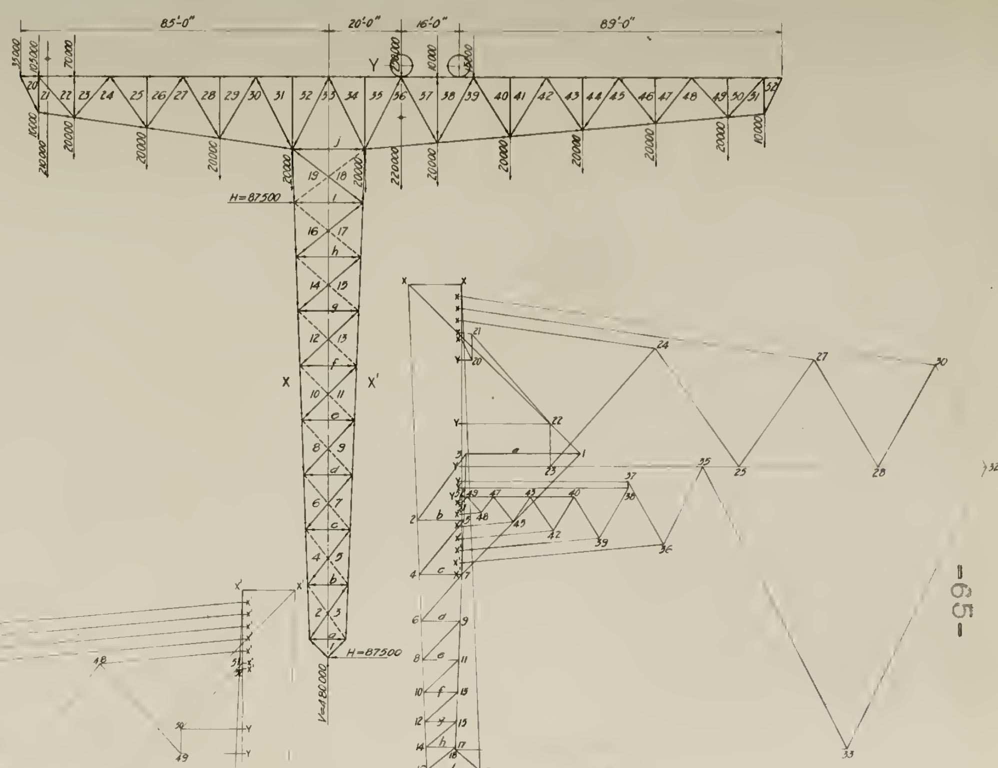
Tower Stresses

Member	Dead Load Stress lb.	Wind Load Stress: Wind Right lb.	Wind Load Stress: Wind Left lb.	Longitudinal Stress: Force Right lb.	Longitudinal Stress: Force Left lb.	Maximum lb.	Minimum lb.
X-1	- 15,000	0	0	- 25,000	+ 25,000	- 40,000	+ 10,000
X-3	- 45,000	- 2,200	+2,200	- 63,000	+ 63,000	-110,000	+ 20,200
X-5	-100,000	- 8,200	+8,200	- 89,000	+ 89,000	-197,000	- 2,800
X'-2	+ 15,000	+ 2,200	-2,200	+ 63,000	- 63,000	+ 80,200	-50,200
X'-4	+ 45,000	+ 8,200	-8,200	+ 89,000	- 89,000	+142,200	-52,200
X'-6	+100,000	+ 1,640	-1,640	+107,000	-107,000	+208,640	- 8,640
X'-1	- 2,000	0	-2,100	+ 35,000	- 35,000	- 39,100	+33,000
2-3	- 5,000	+ 1,600	-6,100	+ 27,000	- 27,000	- 38,100	+23,600
4-5	- 11,000	+ 4,800	-9,800	+ 21,000	- 21,000	- 41,800	+ 14,800
Y-6	0	+ 2,700	-5,300	+ 9,000	- 9,000	- 14,300	+ 11,700
1-2	0	- 2,900	+2,900	- 49,000	+ 49,000	- 51,900	+ 51,900
3-4	0	- 8,100	+8,100	- 35,000	+ 35,000	- 43,100	+ 43,100
5-6	0	-12,000	+12,000	- 26,000	+ 26,000	- 38,000	+ 38,000



Stress Diagram Due to Maximum Loading
Scale 1 in = 80000 lbs

Fig. 36



Stress Diagram Due to Minimum Loading
Scale 1 in. = 80 000 lbs.

Fig. 37

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UNIVERSITY OF ILLINOIS
THESIS
DESIGN OF A 150-TON
ROTARY TOWER CRANE
Scale of Structure 1 in = 40 ft
April 6, 1911

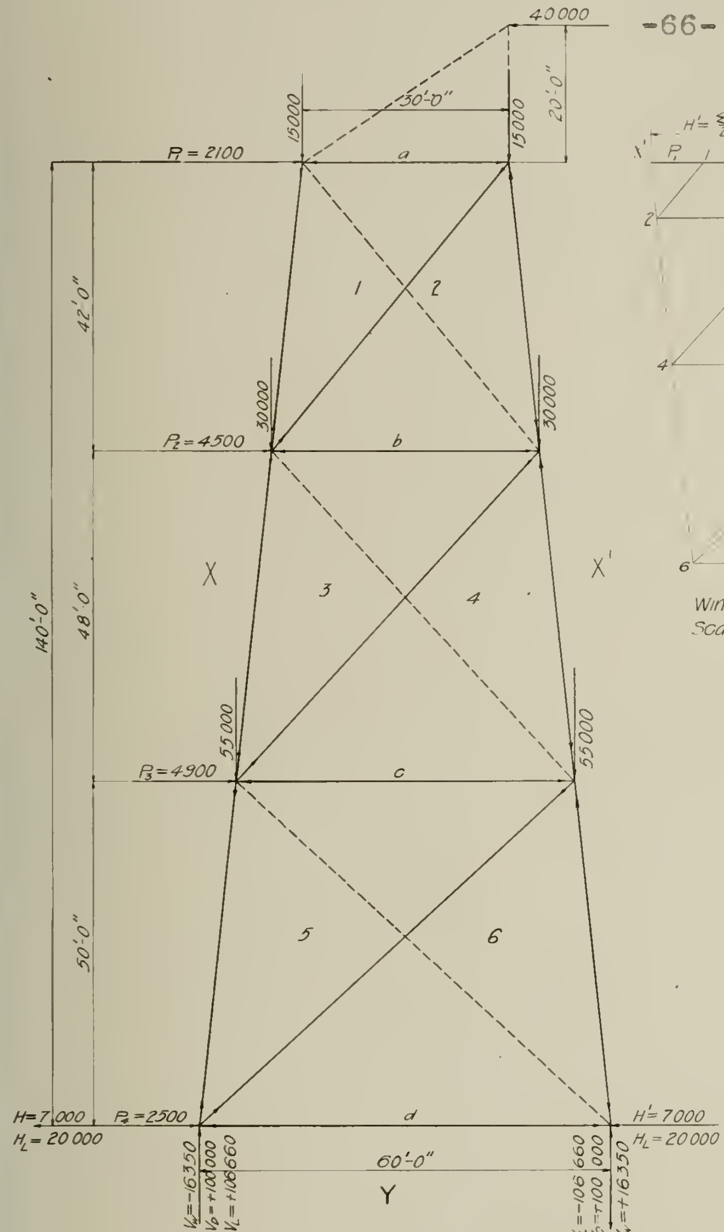
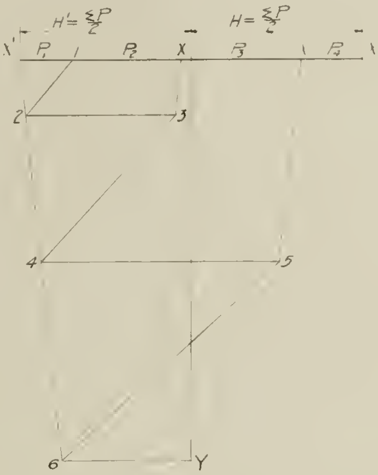
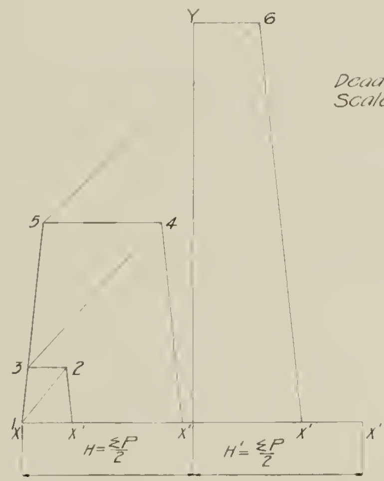


Fig. 38

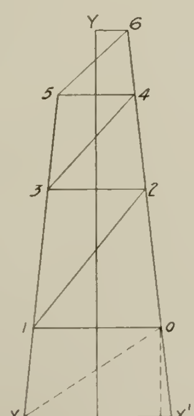


Wind Stresses - Wind Left
Scale: 1 inch = 3000 Pounds
Fig. 39

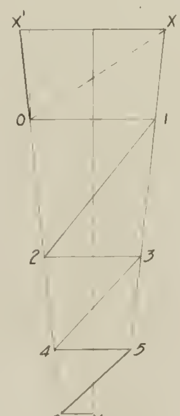
Dead load Stresses
Scale 1 inch = 20000 Pounds
Fig 40



Wind Stresses Wind Right
Scale: 1 inch = 3000 Pounds
Fig 41



Stresses Due to Longitudinal Force (Right)
Scale 1 inch = 20000 Pounds
Fig 42



Force (Left)
Fig 43

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THESIS
DESIGN OF A 150 TON
ROTARY TOWER CRANE
Scale of Structure $\frac{3}{32}$ in = 1 ft
May 24 1911 P F Pofahl